DESIGN OF A BULLDOZER FOR THE LUNAR ENVIRONMENT

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ABSTRACT

In the process of researching the establishment of a base on the lunar surface, N.A.S.A. has recognized the need for soil moving equipment. This equipment is vital to site preparation. A suitable design for a bulldozer capable of operation on the lunar surface is investigated. Design requires that a working knowledge of the lunar environment and its effect upon the operation and product life of such equipment be attained. These factors present special limitations during the design phase and different alternatives are explored.

INTRODUCTION

This report serves as a recommendation to N.A.S.A. in fullfillment of the requirements of M.E.4182 by outlining the design of a bulldozer specially equipped for implementation on the lunar surface.

2.1 Problem Identification: Special atmospheric and geophysical conditions on the earth's moon make existing earth moving equipment unsuitable for this task. An existing earth model, the Case uniloader 1845B, is used as a basis for the lunar design.

2.2 Constraints: In addition to environmental limitations, the performance objectives and operating constraints are factors which are vital to the design. These constraints are primarily based on those of the Case Uniloader. Cost is given minimal consideration due to the fact that this is specialized equipment for limited application rather than a consumer product aimed at competing in a defined market.

2.3. Special Considerations: Design alternatives are researched for optimizing the ease of interfacing man with machine. Human factors of operation are set as design criteria and are given great consideration in the determination of the final design.

3. Problem Statement

The lunar environment greatly differs from that on Earth. These differences pose the engineering challenges surrounding the design of a bulldozer for a lunar application. Design of existing soil moving construction equipment is based upon the environment here on earth. The relevant environmental constraints are: the gravitational field, the mean temperature range, atmospheric conditions, length of daylight, and the mechanical properties of the soil, all of which are different on the Moon.

The size and mass of the Moon and the resultant gravitational field has the most profound effect upon the operating environment to which construction equipment would be exposed. The lunar acceleration of gravity is approximately one sixth that of earth or 5.32 ft/sec^2 on the Moon compared to 32.2 ft/sec^2 on Earth. Thus, an existing design for a bulldozer or loader would weigh much less on the Moon and its ability to maintain traction would be greatly reduced. In addition to weight considerations the lunar gravitational field does not provide a gaseous atmosphere that compares to that of Earth. Without a gaseous atmosphere, convective heat transfer becomes impossible. Thus, conventional fan and "radiator" cooling devices utilized on Earth would be inadequate on the Moon. Further, the lack of atmosphere results in virtually no atmospheric pressure. Thus, exposed, uncontained fluids, such as lubricants, will tend to vaporize.

The moon rotates on its axis once every 27.3 earth days. Daylight lasts for approximately 328 hours at a stretch. Temperatures range from 220°K (-64°F) at sunrise, to 340°K

(155°F) during midday. This data was collected by the Apollo 11 lunar flight at the moon's equator.

In addition, the science of soil mechanics and vehicle mobility has produced relationships (between) soil characteristics that are generally accepted foundations for determining appropriate vehicular forms. NASA became aware of these characteristics most notibly through work conducted by the General Motors Defense Research Laboratories. Strengh values for the cohesion (c) and the friction angle ( ), along with deformation values for friction modulus of deformation (K_0), cohesion modulus (k_c), sinkage exponent (n), and slip coefficient (K) for lunar soil were obtained from the Apollo Program.

The specific/individual effects of all of the preceding environmental characteristics will be discussed in greater detail as the more detailed design considerations are outlined.

4. PERFORMANCE OBJECTIVES

Common in unfamiliar design problems, analysis of existing similar design solutions is often taken into consideration to furnish a starting point in the design process. In designing the lunar bulldozer, the CASE 1835B Uni-Loader was chosen as a design guide to physical and performance specifications. In the course of design, some parameters of the CASE model were modified to suit varying aspects in regard to operation, environment, etc.

4.1. Payload: Payload specifications are derived from the density of lunar soil and the maximum bucket size available for the CASE front-end loading attachment. Using 2.716 lbm/ft^3 as the density of lunar soil and 18.8 ft^3 as the SAE heaped-bucket capacity, a payload of roughly 500 lbf is calculated. It is assumed, however, that the hydraulic system operating the attachment will create a lift capacity many times greater than this value.

4.2. Duty Cycle: Design requires that the period of operation of the lunar bulldozer not exceed six hours due to limitations in heat removal and power supply. The full cycle of operation is contingent upon a replenished coolant and power supply.

4.3. Maintenance: The lunar bulldozer was designed for ease of maintenance (i.e. systems check, service). A rigorous maintenance schedule is difficult to establish due to lack of testing and experimentation; however, problems encountered should be isolated and their causes determined so that corrective measures can be taken. Aside from replenishing supplies, these corrective measures may involve only component replacement or even structure disassembly. It is assumed that persons implementing maintenance procedures understand fully the function of all components and how systems are to be disassembled and assembled.

4.4. Operating Constraints

4.4.1. Speed: The speed is limited by several factors. These include depth and type of soil, vehicle weight, unevenness of ground surface, and drive system specifications. While these factors were taken into consideration, a maximum operating speed of six miles per hour was arbitrarily chosen after investigation of the CASE model maximum speed. This value was also found to be in agreement with the limiting factors mentioned above. For example, with a maximum motor speed of 3500 rpm, 60:1 gear reduction, and 36 inch tires, a vehicle speed of six mph can be obtained.

4.4.2. Dimensions and Weight: While major dimensions initially evolved from investigation of the CASE model, consideration of area and volume requirements for such components as fuel cells, cooling tanks, and drive system components brought about final optimum dimensions. Overall dimensions include length as 140 inches, width as 86 inches, and height as 80 inches. For proper tractive effort, a vehicle weight of roughly 2800 lbf is necessary (lunar weight; 525 lbm in general).

4.4.3. Power/Fuel Comsumption: The fuel tanks are sized to supply fuel for six hours of maximum operation. Assuming the bulldozer will operate on average at 95 percent maximum power, the bulldozer can operate for 7 hours and 21 minutes. For this duty cycle, the fuel cell uses a total volume of 21.45 ft^3 of Hydrogen and 10.72 ft^3 of Oxygen. Both are to be at 3000 psi.

4.4.4. <u>Traction:</u> Early in the design process, it was realized that some determination of vehicle traction requirements and capabilities must be made. It was obvious that the reduced gravitational field would diminish the normal forces upon the tracks or wheels of the bulldozer, resulting in soil shear or "slip" when torque was applied to the driving wheels, if the proper weight and/or contact area was not achieved.

Initial research led to the work of M. G. Bekker in the area of land locomotion mechanics. Bekker outlines in several sources , the equations that can be used to approximate vehicular performance in a soil of known strength and deformation characteristics.

Based upon soil density, cohesion, friction angle, track or wheel contact area, and a specified sinkage, a safe weight that will keep the vehicle at or above the prescribed sinkage can be determined from the safe weight equations (see Appendix 10.3). This equation is coupled with the soil dependent sinkage relationship for a given tractive geometry.

An iterative process is employed, assuming sinkage, tractive width, and effective diameter. Once the correct values for weight, width, and diameter are determined, a determination of the soil thrust, or useful tractive force, and drawbar pull can be made. Drawbar pull is the difference between the soil shear strength (thrust capacity) and the motion resistance of the vehicle. Motion resistance results from soil compaction, wheel or track "bulldozing" due to sinkage, grade, or slope resistance, and inertia resistance during acceleration.

The work of Bekker and others indicates that the motion resistance of a tracked vehicle is approximately three times that of a wheeled vehicle. Since the lunar soil has minimal cohesional properties, the lesser tractive area of a wheel is not the dominant factor in the soil thrust equation. Thus, for our purposes, an assumed sinkage of three inches, a wheel width of fourteen inches, and a diameter of 36 inches was found to be appropriate. The resultant elliptical contact area was found to be 218 square inches per wheel. This leads to a safe weight of nearly 940 pounds per wheel and a weight of 880 pounds per wheel to cause three inch sinkage. Soil thrust was calculated including a one inch tread or grouser depth. For these conditions, a soil thrust capacity of 660 pounds is expected. Compaction resistance and bulldozing resistance were found to be 190 and 150 pounds per wheel, respectively. The resulting drawbar pull is 320 pounds per wheel; this gives a total of nearly 1300 pounds of useful tractive force. The results can be summarized as follows:

Wheel diameter = 36 inches

Wheel width = 14 inches

Total weight = 3520 (lunar) pounds

Drawbar pull = 1300 pounds

5.1.1. **Motors:** The lunar bulldozer will be using fuel cells for power; therefore, the only kind of motor that was considered for the design was the D.C. motor.

From a characteristic and application standpoint, D.C. motors are classified according to their type of field winding. There are three types: shunt wound, series wound, and compound wound. The operating characteristics of these three types differ and are described separately.

MOTOR OPERATING CHARACTERISTICS

Shunt Motors: Shunt motors run at very nearly the same speed at any load within their capacity. There is but a slight drop in speed from no load to full load. This fact makes shunt motors suitable for driving machinery that is designed to run continuously at a constant speed. The most important limitation in the use of a shunt motor is its moderate starting torque.

Series Motors: Series motors are suitable in load applications where it is necessary to supply a large torque with a moderate increase in current, such as in traction work, crane operation, etc. The speed of a series motor varies greatly with the change in load. Because of this speed characteristic, and the resultant possibility of dangerously high speed at light loads, this motor is not suitable for belt drive or for use on any load where the torque might drop below 15% of full-load torque. Another limitation is that the speed range cannot be adjusted by practical means. In this regard, the series motor cannot be considered a flexible motor.

Compound Motors: The addition of a cumulative series field winding to the shunt field produces a compound motor. The addition of this series-field winding gives the motor a characteristic which is a combination of the series and shunt motor.

In compound motors, the speed changes with the load, but it does not change as much as in a series motor. On the other hand, the speed changes a great deal more than in a shunt motor. Compound motors are used for loads requiring high starting torque, or for loads subject to torque pulsations. They are employed for elevators, air compressors, printing presses, etc.

TEMPERATURE RISE AND INSULATION

The temperature of an electric motor has an adverse effect on the life of the motor winding. Types of insulation required for various temperature ranges, and methods for avoiding high temperatures are discussed.

NEMA defines the following insulation classes: class A, class B, class F, and class H. The temperature limitations of the four classes are as follows:

INSULATION CLASS	MAXIMUM TEMPERATURE (C)
A	105
B	130
F	155
H	180

Temperatures shown next to each insulation class can be interpreted as follows: From the temperature shown, subtract 40 °C for ambient allowance, and 10 °C for hot-spot allowance (hot-spot is the hottest part of the winding that will fail first). This leaves the total allowable rise at which the motor will run safely.

The life of the motor is inversely proportional to the temperature of the windings. It is therefore possible to increase the motor life by reducing the temperature of the windings. This can be accomplished by selecting one of the following alternatives:

1. Select an enclosure or insulation treatment to combat environmental conditions.
2. Select high class insulations (F or H).
3. Select a high HP rating.
4. Select a multispeed or wound rotor motor, for frequent or high inertia starting.

The use of high-temperature insulating materials in electric motors has opened up possibilities that are not yet entirely exploited. The applicability of the various classes of insulation are described below:

Class A: This class consists of organic materials for which normal life can be expected if the insulation is operated below 105°C.

Class B: This class includes glass fiber, asbestos and mica for which normal insulation life can be obtained if final temperature does not exceed 130 °C.

Class F: Extensive research with various plastic films and varnishes has produced a line of insulating materials for intermediate temperature ranges. These can be used in motors operating up to a total temperature of 155 °C.

Class H: With the development of silicones for insulation impregnation, new high-temperature standards became practicable. In such designs, the final operating temperature of the insulation system can be 180 °C.

ENCLOSURES

The various types of enclosures used for protection of the motors are reflected in the NEMA standardized motor types. Specifically, these enclosure types are designated by the degree of protection that the motor is afforded. Some kinds of enclosures are the following:

TENV (Totally enclosed non-ventilated)

TEFV (Totally enclosed force-ventilated)

Explosion proof

TEWC (Totally enclosed water-cooled)

The enclosure that is of concern to the design is the TEWC. In this kind of enclosure, the motors are cooled by circulating water that is in direct contact with the motor parts.

MOUNTINGS

Methods of mounting have been specified and standardized by NEMA. These include rigid mountings, resilient mountings, and flange or face mountings.

Rigid Mounting: This is the simplest and least expensive mothod of mounting a motor. The position of the base mounting holes with reference to each other and to the location of the shaft is specified by NEMA.

Resilient Mountings: This method isolates motor vibrations and reduces noise. The most effective arrangement is to introduce the resiliency as close as possible to the hubs.

Flange or Face Mountings: NEMA has standardized two end mounts, types C and D, and two flange mounts, types P and PH.

MOTOR SELECTION

The space dozer will be using the following motors for drive, hydraulic pump and cooling system.

	DRIVE SYSTEM	HYDRAULIC PUMP	COOLING SYSTEM
Number of motors	2	1	1
HP (kw)	20 (15)	7 (5)	0.5 (0.37)
Winding	series	series	series
RPM	3500	2000	1200
Torque (max.) in in-lbf.	360	220	26
Full load current (amps)	63	25	2
Enclosure	TEWC	TEWC	TEWC
Insulation class	H	H	H

5.1.2. DRIVETRAINThe purpose of the drivetrain is to provide necessary gear reduction and torque multiplication and possible reversing capability. Selection of a suitable system for the lunar bulldozer required careful consideration of the following factors:

(1) High torque capacity (large gear reduction);

(2) High operating efficiency;

(3) Minimum number of moving (heat generating) components;

(4) Compact size or arrangement of components.

Six separate component systems were incorporated in the lunar bulldozer design to accomodate these factors. These include two each of a gear reducer, reversing unit, and final drive unit.

5.1.2.1. Gear Reducer: Because other factors are essentially derived from the specifications chosen for gear reduction and torque capacity, this became the deciding factor in selecting a suitable gear reducer. Problems associated with gear reduction systems are as follows:

(1) Complexity: while there is no limit to the reduction ratio that can be achieved using gearing, arrangements of the components can become quite complex for higher ratios. In a simple gear mesh, a maximum ratio in the order of only 8:1 or 10:1 can be achieved. To achieve higher ratios, planetary, planocentric, or harmonic drive systems can be utilized. These systems feature multiple load sharing paths; thus, size can be reduced significantly.

(2) Dynamic Effects: the operating speed of gears has a significant effect on the design definition. At high speeds, component discrepancies such as tooth spacing error, shafting imbalance, and

so on, generate significant dynamic loading. For these reasons, high-speed components must be of high accuracy to minimize problems; however, this is in many cases difficult to accomplish. To eliminate dynamic loading problems, planocentric and harmonic drive systems can be utilized.

(3) Wear: difficulties arise from the usual point or line contact between teeth in mesh, and from the widely varying rates of relative motion of teeth during engagement. Individual teeth are also frequently worn more rapidly than other teeth. Aggravations occur during impact loading, when an impact load may be encountered at a time when particular teeth are in engagement and carrying the entire load. Impact effects are likely to cause brinelling, heating, cracking, galling, and accelerated failure. As stated before, planetary, planocentric, and harmonic drive systems feature multiple load sharing paths and can be utilized to alleviate problems with wear.

Following evaluations of various gear reduction systems, a harmonic unit with 60:1 reduction was chosen. The 60:1 ratio was fixed from motor and vehicle speed specification. The radically different principles upon which the operation of harmonic drive depends produces parameters differing considerably from those for conventional gearing. Harmonic drive depends on the elastic deflection of one or more of its components, specifically the strain gear. One complete revolution of the strain inducer (input shaft) will always produce an output tooth movement which is equal to the difference in the number of teeth between ring gear and strain gear.

Thus, high gear ratios are possible. Other advantages are noteworthy and include the following:

(1) High contact ratio (at least 50 percent of teeth in contact at all times), resulting in high torque capacities;

(2) Spline teeth come into contact with an almost pure radial motion and have essentially zero sliding velocity, even at high input speeds. Tooth friction losses and tooth wear are very low;

(3) Because of the low friction losses, mechanical efficiencies to the extent of 95 percent can be obtained. Unlike conventional gearing, efficiencies in harmonic drives are particularly outstanding at high ratios;

(4) Spline teeth in contact and under load are practically stationary. Under normal operating conditions, dynamic loading is very low;

(5) Regions of tooth engagement and application of load torque are usually diametrically opposed and result in counterbalanced forces. Therefore, the bearings used in a harmonic drive system are only to enclose the mechanism and withstand externally applied loads;

(6) Tubular construction of splined elements results in high torsional rigidity and thus less windup under load;

(7) Due to the high torque capacity, harmonic drive units can be made relatively very small in size as compared to conventional gearing units. Unfortunately, as a consequence of small size and light weight, the thermal capacity of a harmonic drive unit will in most cases be its limiting characteristic. In order to realize its potential capacity, forced cooling means should be used for continous heavy-duty service.

5.1.2.2. Reversing Unit: The reversing unit was designed to accomodate the high output torque from the harmonic drive unit and provide for the following modes of operation: forward, neutral, and reverse. This type of design was originally adapted from equivalent fractional-horsepower gearbox designs in applications such as riding lawnmowers.

Forward mode is accomplished by the positive-drive ratio of a silent chain drive. Silent chain was chosen for the following reasons:

(1) Smooth and efficient operation even in rigorous applications;

(2) Suitable for transmission of power over a wide range of loads and speeds;

(3) Especially suitable for short drive-centers, small diameter sprockets, and low speeds.

Reverse mode is accomplished by a spur gear drive (providing the negative-drive ratio). Despite the advantages of helical gearing in regard to noise and vibration, the resulting generation of an axial thrust would necessitate the addition of a thrust bearing on each helical gear shaft and add complexity to the system. The use of spur gearing was felt adequate in this case.

Drive mode (in contrast to "neutral") is accessed by engaging the fixed spline on the input shaft (essentially the harmonic drive output shaft) with either of the free-spinning sprocket or spur gear splines by means of and internal spline coupler. Thus, output to the final drive unit is the result of power diverted through the silent chain drive or spur gear drive.

5.1.2.3. Final Drive Unit: The final drive unit is responsible for diverting power output from the reversing unit to the drive wheels.

Utilization of the skid-steering concept required that both front and rear drive wheels on either side of the vehicle rotate in unison. Due to the greater center distance of 60 inches between front and rear drive axles, consideration was given to the following for effective transmission of power: belt drives, chain drives, and bevel gear drives (a miter gearbox at each axle with a connecting shaft).

Because of the inherent slippage in belt drives and complexity of the bevel drives, a roller chain drive is recommended for this application. To transmit maximum power, two double-strand ANSI #100 roller chains with four twin 26-tooth sprockets are required. In addition, another set with shorter center distance is requred to accomodate the physical arrangement between reversing unit output and rear drive axles.

5.1.3. **Wheels:** SAE Standards for Tractor/Implement disc wheels offer a wide range of sizes and strength ratings. Pertinent dimensions and mounting requirements are easily specified. For the requirements of this project, a wheel, SAE #806, was chosen. The diameter is 18 inches with a 14 inch rim width. The radial load rating is 5000 pounds; this exceeds our load requirements, but was chosen due to its dimensions and the projected ease with which it might be altered or modified to meet our needs. We require a 36 inch diameter tire, which would normally be met with a pnuematic tire. However, a solid wheel constructed of wire mesh , as for the LRV used during the Apollo missions, or a spoked metal rim addition with angle iron "treads" as used in early tractor designs would be most suitable for our needs.

5.1.4. **Bearings:** In selecting bearings for a given application, consideration must be given to the following:

(a). specified dimensions of the space where the bearings are to be mounted.

(b). loads the bearing will need to withstand

(c). satisfactory life required of the bearings under specified operating conditions.

Due to the fact that thrust loads will be generated at the final drive axis, the use of tapered roller bearings is recommended. Since they can withstand a combination of radial and thrust loads, tapered roller bearings combine the advantages of ball and straight roller bearings. In addition, tapered roller bearings are especially suited for low-speed operation.

Each shaft will require two tapered roller bearings in direct mounting to cancel the self-induced thrust reactions caused by taper. A total of eight bearings is required for the drive axles: four 2.25 inch bore diameter tapered roller bearings with a width of 1.50 inches, an outer diameter of 4.875 inches, a radial load rating of 9,280 lbf and an axial rating of 5490 lbf (Timken part no. TS 555-S cup and TS 552A cone); four 2.50 inch bore diameter two-row tapered roller bearings with an outer diameter of 4.72 inches, a width of 2.812 inches, a radial load rating of 18,900 lbf and a thrust rating of 6,260 lbf (Timken part no. JH307749-90N02, Type SR two-row bearings). [For calculations of the required load ratings for the above bearings, please see Appendix 10.3.].

5.2. Cooling System: Thermal control of the lunar dozer is required to maintain components within prescribed temperature limits.

A thermal system similar to that of the Lunar Roving Vehicle was proposed. This system called for passive thermal control techniques consisting of selected radiation surface finishes, heat sinks, flexible thermal straps, multilayer insulation, and low thermal conductance component mounts. All systems would be on board the vehicle. This seemed like a feasible system until the total heat dissipation of all the mechanical parts was calculated. The heat dissipation from each component is shown below:

PART	HEAT DISSIPATION (rough est.)
L. D. motor	2 kw
R. D. motor	2 kw
H. P. motor	2 kw
C. S. motor	1 kw
Fuel cells	11 kw
TOTAL:	18 kw

where

L. D. - left drive

R. D. - right drive

H. P. - hydraulic pump

C. S. - cooling system

Since, on the moon there is no convective heat transfer, the heat should be removed by either radiation or by storing it into large heat sinks with subsequent radiation to space. We would need an optical solar reflector with an area of 20.50 m^2 to get rid of the heat (see cooling system calculations in Appendix 10.3.). This is impractical, so the idea of rejecting the head via a solar reflector was abandoned.

Actual Cooling System: We estimate the heat rejection on all heat-dissipating components to be 108 hw-h (18 kw for 6 hrs.). Using the thermodynamic relationship, $Q = mc_p(\ T)$, we figured that, using 320 gallons of water, we could accomodate all the energy with a temperature rise in the water of 80 •C.

The devised cooling system works as following:

General Operation: As soon as the system is turned on, valves v_{11}, v_{21}, and v_{31} (see cooling schematic in Appendix) are all open; valve v_{41} is closed and transfer valve (k) is set in k_3 position. Valves v_{12}, v_{22}, and v_{32} are all closed to prevent contact with a vacuum. Pump P1 is turned on. The water flows from tank 3 through the pump; it then splits and enters 4 different pipe systems. Each pipe system is connected to the drive motor 1, the drive motor 2, the hydraulic motor and the fuel cell heat exchanger, respectively. The water, after leaving the four different pipes, converges to a single pipe and is then poured in tank 4.

Subsystem A: This system contains a volumetric flow rate measuring device (M_A) that keeps track of the amount of water that enters tank 4. It also contains a valve, v_{41}, whose significance will be apparent when subsystem B is analyzed.

Subsystem B: When M_A reads 100 gallons, this means that tank 3 is almost empty. At that instant, valve v_{31} closes and valve v_{32} is opened instantaneously in order to introduce a vacuum into tank 3. Then valve v_{41} opens, thus allowing all the hot water from tank 4 to flow to tank 3. No pump is necessary for this due to the pressure difference between tanks 3 and 4 that "drives" the water from tank 4 to tank 3. After all water has flown to tank 3, valve 41 closes and the transfer valve is switched to the k_2 position. When the meter reads 100 gallons, the same procedure occurs between tanks 4 and 2.

<u>Subsystem C:</u> This system contains the pump P1 that is the heart of the entire system. The pump is controlled in order to maintain a constant pressure on its output side.

<u>Subsystem D:</u> In this system, all the heat transfer occurs. The pipe initially splits into 4 different pipes, each connected to the drive motor 1, drive motor 2, the hydraulic motor and the fuel cells. Each of these pipes contains a valve followed by the actual component and a temperature-measuring device, which is in turn connected to the valve through a control system. The desirable temperature differences for the water before and after passing through the component is 80°C. The control system will measure the temperature at the outlet and send an appropriate signal to the valve. If the outlet temperature is less than 85 degrees, then the valve is shut so that the flow rate through the motor is decreased; therefore, the exit temperature is increased (and vice versa).

<u>Heat Rejection from Water Tanks:</u> The dozer will operate for 6 hours; it will be parked for 18 hours until the next 6 hour duty cycle. During the 18 hours that the dozer is not operating, the heat sink must get rid of all the stored heat, so that the system will be ready for the next operating period.

The water temperature must decrease from 85°C to 5°C during those 18 hours. Since the rate of heat radiation depends heavily on temperature (~T), heat will be rejected very quickly at the beginning, leading to a high sink temperature decrease, and slower towards the end, leading to low sink temperature deceases.

Using the HP41C, iterations were performed with respect to the area A that is needed to reduce the temperature from 85°C to 5°C. Since the temperature changes all the time, a constant sink temperature during a one hour interval was assumed (for results, see calculations in Appendix 10.3.). It was determined that a solar radiator 15 m^2 in area was needed. The heat sink will cool down to 5°C in about 16 hours, thus allowing for two hours between transportation of the heat sink from the vehicle to the solar radiator reflectors and vice-versa (the actual time is 16 hours, 19 minutes).

The solar reflectors will be the same kind that were used with the lunar rover. They are second surface mirrors made by vacuum deposition of a silver film on a thin, fused silica wafer. The surfaces are protected from the atmosphere by overcoating the silver with vacuum deposited inconel. The reflectors will be covered by insulating dust covers that are located over the radiator to prevent undesirable cool-down and accumulations of lunar dust which will degrade the radiative surface properties of the reflectors. The covers will be opened manually by the astronaut, byt they will shut automatically, as soon as the temperature of the sink reaches 5°C (This system was used with success in the Lunar Roving Vehicle).

Water Tanks: Three 0.45m tanks measuring 1m x 1m x 0.45m will each carry 105.6 gallons of water. The faces of the tanks will be made of 6061 -T6 aluminum plates 3.2mm thick and will be welded together. In addition, the tanks will be covered by an insulation blanket consisting of 15 layers of perforated double aluminized mylar, assembled by heat sealing the edges using silicone adhesive particles (0.25" squares on 1.5" centers, staggered from layer to layer). The insulation will minimize any heat exchange between the tanks and the relatively severe lunar surface environment.

5.3 STRUCTURAL SUPPORTS:

5.3.1 CHASSIS:

The chassis of the bulldozer is a combination of a steel bar frame with 1 layer of welded stainless steel on hte outside and on the inside. The insulation outlined in section 5.22 is contained between the two plates. The bottom has only one plate which extends out past the sides to provide support for the wheel mounts.This design provides total containment of the internal cavity from the lunar dust and provides added structural support for the mounting of the fuel cell and fuel tanks on the inside as well as mounting the lift arm assembly and wheel mounts on the outside.

The bottom of the chassis also extends 2 ft in the rear to provide support for the drive motors. This bottom plate is 1/2 inch thick and is stainless steel.

The other plates on the chassis are 1/4 inch thick stainless steel.

This type of construction was selected for its low thermal conductivity.

The top plate is cut and hinged 3 feet from the back edge of the chassis box to allow access to the fuel cells and fuel tanks.

The steel reinforcement bars are 1/4 inch I-Beams. They are welded together and the steel plates are welded to them. They also prevent the insulation from moving between the plates.

Because design calculations of a combined strucutre is difficult to perform, a superposition procedure is used to guarantee the needed strength of the chassis.

5.3.2 LIFT ARM ASSEMBLY MOUNTS:

Two important considerations, total moment and total shear forces, are taken into account in designing a support for the lift arms. First, the 2 mounts must support the total weight of the soil in the bucket. Because the soil on the moon is less dense than the soil on the earth, and because of the gravity difference, the total lift forces would be 500 lbf under maximum conditions. This translates into a total shear force of 250 lb which the metal and its welds must withstand.

Second, the 2 mounts must be welded so that hte joints can substain the 5200 in lbf experienced during bulldozing as a result of the 1300lbf drawbar pull force. Obviously, the second condition has higher stresses and if this criteria is satisfied, the first will be satisfied as well.

Based on these forces and the design ofpattern, 1/8 inch steel plates were selected as the size necessary to support the arms.

5.3.3. Wheel mounts: The wheels will be mounted to the chassis by 2 support eye members. because the entire shaft will rotate, bearings will be press fit into each eye. One eye will be against the wall of the chassis and the other will be on the outmost part of the running board.

They both will be bolted to the chassis. In addition, a metal casing will surround this entire chain section to protect it from the dust.

5.4 POWER SUPPLY

In this section of the design, the viable alternatives were fuel cells and storage batteries. Because storage bateries occupy a large volume just to supply the bulldozer with ample power, fuel cells were adopted as the source of power. In addition, storage batteries require a slow regineration at low voltages. This would not allow for adequate battery turnaround, thus reinforcing the decision to use fuel cells.

5.4.1 FUEL CELL TYPE: A hydrogen oxygen fule cell was selected primarly because of two reasons. One, the by-product, water, could be separated by an electrolysis device powered by solar solar energy.

Second, additional fuel could be transported on sight in the form of water without the risk of possible explosion of gases. This water could be transported form the earth or from the lunar polar caps. Fuel cells require miminal maintenance and can be started up instantaneously at temperatures as low as -10°F.

5.4.2 THERMOCHEMISTRY BACKGROUND: Power is produced in a hydrogen oxygen fuel cell by the release of a free electron to an external circuit during oxidation of a hydrogen molecule. This process is achieved by supplying hydrogen and oxygen to two seperate electrodes immersed in a electrlytic solution. The free electrons migrate to the anode and the active hydrogen ion travels through the electrolytes to combine with the oxygen ion to form water. Usually, the electrolyte is a potassium hydroxide of sulfuric solution. The chemical reaction between the hydrogen and oxygen molecules is an equilibrium reaction. This reaction allows the total system to achieve a lower energy state by forming a more stable compound, water, and releasing the extra electrons of the relatively unstable hydrogen molecule. The empirical reaction for the oxidation of hydrogen in an alkaline electrolyte is:

$$2H_2 + 2O_2 \quad 2H_2 + 2e^-$$

5.4.3. **DESIGN METHOD:** In designing a fuel cell for the NASA lunar bulldozer, a linear extrapolation of available data based on relative KWH demands can provide an approximation and a ballpark figure for the new design. This method is used to project the size and weight specifications of the fuel cell for the NASA bulldozer.

5.4.4. **PROJECTED SSPECIFICATIONS:** For a total demand of 240KWH, a hydrogen -oxygen fuel cell operating at 85% efficiency has the following projected specifications.

Weight of fuel cell	1600 lbf
Volume of fuel cell	20 ft^3
Hydrogen required, per KWH at STP	108 ft^3
Oxygen required, per KWH at STP	54 ft^3
Volume of O_2 andH_2 at 3000psi	32.2 ft^3
Weight of O_2 and H_2 and fuel tanks	450.0 lbf
Total weight of fuel cell and fuel system	2050 lbf

5.4.5. **EVALUATION:** Based on this extrapolation, the hydrogen oxygen fuel cell can power the NASA bulldozer at peak power continuously over a 6 hour duty cycle. In addition, the fuel cell and the fuel tanks meet the size and weight limitations of the project. Thermal insulation and water cooling will provide adequate heat removal to maintain a constant operating temperaturer in the cell.

5.4.6. TANK SPECIFICATIONS: The fuel tanks for the fule cell must meet the following specifications:

	H_2 TANK	O_2 TANK
Volume	21.45 ft^3	10.72 ft^3
Dimensions	3'x4.25'x1.7'	1.1'x3'x4.25'
Pressure	3000 PSI MAX	3000 PSI MAX

The tanks must be internally braced and constructed with rounded corners.

5.5. <u>Insulation Materials and Coatings:</u> There are three dangers to materials in space: ultraviolet radiation, particle radiation and micrometeorite rain. Since shielding the astronaut is not a primary consideration in this design (the dozer is operated by remote control), metallic coatings are recommended. For thermal protection, all exterior surfaces will be painted a bright white. A recommended coating is <u>Chemical Fabrics</u> by Birdair Structures of Buffalo, New York; this coating has a fiberglass base with a teflon exterior surface. Originally a soft, light brown film, this material reacts with the sun in seconds to form a hard, brilliant white coating. For protection against micrometeorite pitting, it is also suggested that several thin layers of Dupont Kevlar K-29 (a textile fiber composite) and neoprene rubber be used directly under the metallic coating.

G-10 fiberglass is the recommended insulation material. Even a relatively small puncture in one of the components would cause loose fiberglass insulation to escape and float around in space; therefore, the fiberglass should be encased in several layers of parachute cloth or a similar filter fabric. By lining the chassis with a layer of neoprene rubber in addition to the fiberglass insulation, we can also minimize the shock caused by sudden stops.

6.1. Ergonomics/Controls: After the volume requirements of the powerplant, the fuel tanks, and the cooling system became apparent, we decided upon remote control for the operation of the dozer. Remote operation has a number of advantages. First, the operator will not run the risk of suit puncture by the sharp edges or ridges that may arise during the normal operation of the dozer. For example, extended operation of the dozer in rocky areas could wear the lower edges of the bumpers and fenders until these edges were sharp enough to puncture the operator's suit.

Also, the operator will not have to climb over the lift arms or other parts of the dozer in order to reach the control panel. The severe motion restrictions imposed by the operator's spacesuit will therefore not pose a problem. By controlling the dozer from the outside, the operator will also enjoy a better view of the area surrounding the dozer. This should lessen the likelyhood of backing or turning into obstacles that might otherwise be blocked from the operator's view.

The operator may control the dozer from a small vehicle similar to the Lunar Roving Vehicle used in various Apollo Missions. This would enable the operator to return to base during his work shift without using the relatively large quanity of fuel it would require to drive the dozer to base and back to the worksite. The smaller vehicle could also be used to take the operator to base, should the dozer break down on the worksite.

There are also safety advantages that arise from the remote operation of the dozer. For instance, if the dozer should overturn, the operator's life would not be jeopardized. The same is true for the case of fire. Hydrogen and oxygen leakage could lead to a fire, possibly even an explosion, but with remote operation the risk to human life is minimised.

CONTROLS:

The remote controls for the dozer will be located in a self-contained portable control panel. This panel can be mounted to the operator's spacesuit by means of two small battery pods which slide into slots in the front of the suit. (see drawing #, Appendix---). The panel is adjustable for height and angle to suit the operator's preference. Control of the dozer's movements is effected by two control sticks mounted on the panel. All signals transmitted from the panel are received and processed by a computer located on the dozer.

Movement of the contrl sticks affect the dozer as follows. Forward movement of each stick increases the power delivered to that corresponding drive motor (left or right) and the gearbox will be set in forward gear. Rearward movement of the sticks increases the retarding force delivered to the wheels. If one of the sticks is moved rearward while the dozer is moving forward, the computer will initially use the breaking torque of the corresponding to apply the retarding force. If the demanded retardation is greater than the maximum available braking torque of the motor, the computer will actuate the brakes as necessary to provide the desired retardating force. If the operator continues to hold the stick rearward after the dozer has stopped, the computer will engage reverse gear and apply power to the corresponding motor as desired.

Steering is accomplished by contolling the relative speeds of the motors. Movement of the hydraulic cylinders is also controlled by the two sticks. Moving the right-hand (RH) stick <u>away</u> from the center of the panel will lift the entire bucket assembly. Lowering of the bucket assembly is effected by moving the RH stick <u>towards</u> the center of the panel. The left-hand (LH) stick controls the tilt cylinders in much the same way ; moving the LH stick away from the center of the panel will rotate the bucket upwards and moving the stick towards the center of the panel will rotate the bucket downwards. In the event of brake failure, depressing the two buttons atop the control sticks simultaneously (with the thumbs) while rotating both sticks inward will release the emergency brakes and bring the dozer to a halt. The emergency brakes are spring-loaded and must be reset manually by means of two levers, one on each side of the chassis. This feature enables the emergency brakes to function as parking brakes while the dozer is not in service.

In the event of an outwardly obvious fire, the operator may depress the FIRE button on the control panel. Initial depression of this button will cause the computer to perform a rapid temperature check at several key locations inside the dozer. An audible alarm will also be activated. If the computer detects dangerously high temperatures in any spot, it will automatically trigger the on-board fire extinguishing system, which will deliver an inert gas to the problem area via a valve arrangement. If the FIRE button is depressed 3 times within 10 seconds, the on-board extinguisher will deliver a charge of gas throughout the interior of the dozer regardless of the computer's diagnosis.

The dozer's computer also provides several automatic control functions. For example, the computer will keep track of the angle of the lift arm, and the hydraulic fluid flow rate to the lift cylinder. From this information it will determine the vertical speed of the bucket and make the necessary corrections to the hydraulic fluid flow rate to ensure that the lift arm's downward acceleration does not exceed the moon's gravitational acceleration. The computer will also keep track of the dozer's angle in the pitch plane to try to prevent forward or backward somersaulting as a result of applying too much power while pointing up a steep incline or too much braking force whilee travelling down a steep grade. The computer can also be set to disallow wheel locking under braking or wheelspin under power. Furthermore, the computer will not permit reverse gear to be engaged while the dozer is moving forward or vice versa.

One of the primary functions of the computer, however, is to diagnose the dozer's operating condition and control vital systems (such as the cooling system) . Because of the high daytime temperatures experienced near the lunar equator, and because the lack of an atmosphere precludes convective cooling systems, maintaining proper operation of the cooling system is of utmost importance. The computer must read the operating temperatures of the fuel cell battery, the gearbox, each of the dozer's 8 electric motors, and the electronic components of the computer itself, and decide how much cooling water to send to each component to keep it at its optimum operating temperature

Based on both a short-term (2 min.) and a long-term (30 min.) time average of the total demand on the cooling systems and the powerplants, the computer then estimates the time remaining before refueling, replenishing the cooling tanks, or both. This estimate is displayed continuously on the control panel. The fuel pressure is also displayed on the panel in terms of lighted bars; 7 green bars indicate maximum pressure and 1 red bar indicates minimum operating pressure.

The computer wil also display information that the operator requests by depressing a button corresponding to the information he needs to know. Pressing the button a second time will clear the information from the screen. If the computer detects a malfunction, it will flash the DANGER or CAUTION lights on the panel (depending on the severity of the malfunction) and trigger an audible alarm from a speaker inside the operator's helmet. The operator has the option of cancelling the audible or visual signals, or both, by using the two cancellation buttons on the control panel.

If more than one malfunction occurs at one time, the computer will set off both alarms, and display up to three diagnostic messages at once, arranging them in descending order of importance. The control panel also includes a test button to check the circuits in the panel. The panel will be made of a fiberglass-reinforced epoxy laminate, and coated with a white thermal paint. In case of panel malfunction, the dozer can be controlled by an operator at the base.

7. HAZARD ANALYSIS

7.1 Failure Modes: Three failure modes and their consequences are examined in this section: vehicle tip-over, abrupt stops, and explosions. Hazards due to operator misuse are also analyzed; possible operational hazards are attempting to exceed 1) the defined duty cycle, and 2) prescribed performance specifications.

7.1.1. Vehicle Tip-over: The consequences of vehicle tip-over are minimized by the proposed remote-control panel. Since remote-control operation of the dozer allows the operator to control the vehicle from a safe distance, harm to the operator during vehicle tip-over is eliminated. Also, the computer used in the remote control operation of the dozer continuously monitors the maximum angle of tilt that can be attained before the loss of stability occurs. The microprocessor also includes a sensor that shuts off the motor before this angle is reached; therefore, any attempt to travel over terrain which would cause the dozer to tip will cause the vehicle to stop. However, in the unlikely event that vehicle tip-over does occur, the computer is programmed to cease all functions to minimize damage to the vehicle.

7.1.2. Abrupt Stops -- Impact Shock: The second mode of failure analyzed was the effect of abrupt stops on the vehicle. Several steps have been taken to minimize the effects of impact shock on the dozer: the motor is equipped with a special soft-start,soft-stop feature that increases the loading of the shafts gradually; the chassis is also lined with neoprine rubber and fiberglass insulation to protect the fuel cells, gas tanks, and cooling system from punctures that might be caused by sudden stops. The soft-start/soft-stop motor is fairly common is existing earth-moving equipment. Its major disadvantage is that it is not very effective for short, intermittent operations; since the lunar dozer is expected to operate continuously during its six hour duty cycle, this problem should not apply to this design. Since puctures in the gas tanks and cooling system would cause a resulting loss in pressure that would halt the effective operation of these systems, the chassis is lined to protect the above systems from any sharp edges or protruding objects that might cause puncture during sudden stops. The fuel cells are also insulated in a similar manner to minimize the occurrence of explosions due to abrupt stops.

7.1.3. <u>Explosions:</u> Explosions within the lunar dozer were the third inherent mode of failure in this design. Since the astronaut is controlling the operation of the dozer from a safe distance (unless he chooses to ride on the dozer during operation), personal harm due to an explosion is minimized (see Paragraph 1 of this section). The occurrence of an explosin is also minimized because of precautions taken to protect the fuel cells (Paragraph 2). the remaining probable cause of explosion is the refuelling of the cells; in order to minimize damage to the dozer, this will be done at a remote, controlled site. Rubber hoses will be used to refill the tanks (to minimize metal-to-metal contact which may cause sparks in the presence of the oxygen).

7.2 <u>Operator Misuse:</u> Two possible forms of operator misuse were discussed: attempting to exceed the defined duty cycle and attempting to exceed specified performance objectives.

7.2.1. <u>Exceeding the Duty Cycle:</u> Operational hazards caused by attempting to exceed the described duty cycle were minimized, if not eliminated by transducers controlled by the computer. A warning message will alert the astronaut one hour before the proposed termination of the duty cycle (i.e., five hours after operation of the dozer has commenced). If the operator still attempts to exceed six hours of operation, the computer will command the motors to stop.

7.2.2. Exceeding Performance Specifications: The computer will protect the dozer from attempts to exceed performance objectives in a similar manner. Transducers connected to the computer will "weigh" the payload before attempting to lift it; if the payload exceeds predetermined design constraints, the computer will not allow the hydraulic system to operate.

Operating Instructions: Fuel may be added at filling points located on the chassis between the wheels on either side. One filling point is for oxygen and the point on the other side is for filling the hydrogen tanks. To start up the fuel cell battery, the mechanics must then fill the cell with electrolyte before switching the cells on. At that point, hydraulic oil and coolant is added to the system and the pumps started to prime the system. Next, the drive motor circuits are closed, the emergency brakes (one on each side) are set, the remainder of the electronics are switched on, and the dozer is ready to go to work.

Caution must be exercised when powering up the fuel cell battery because of the potentially fatal voltage and high current-delivery capabilities of the powerplant. Coolant should have a temperature of 1°C – 5°C before startup, and the "exhaust" water tank should be empty.

Before full-service operation, the dozer should be given a "pre-run" to test the brakes and the hydraulic system. In normal operation, the computer's warnings should be heeded, especially in the cases of imminent tip-over and overheating. All insulated compartments should be sealed and all chains and bearings properly lubricated before beginning soil-moving operations.

9. CONCLUSIONS AND RECOMMENDATIONS

To meet the specifications required by NASA for a bulldozer suitable for lunar applications, we modified an existing Case #1845B Uni-loader so that it would bulldoze effectively on the moon. Remote control was proposed as a method of operating the dozer; this was intended to minimize harm to the operator and to solve the size constraint problems associated with direct operator access. A thermal system was designed that would effectively combat the problems caused by the lack of convective heat transfer, and thermal coatings and insulation was added to protect the dozer from harm due to radiation and micrometeorite rain.

To prolong the useful life of the dozer, the following "daily" maintenance checklist is recommended:

1. Check chain tension.
2. Check drive train seals.
3. Clean coolant tanks (of dust, etc.)
4. Flush fuel cell battery.
5. Lubricate bearings as necessary.
6. Check on fire retardant pressure.
7. Run circuit checks on computer.
8. Check hydraulic oil level.
9. Change all filters (hydraulic, coolant, and electric) as necessary.
10. Check operation of both emergency brakes.
11. Check wheel condition.
12. Preheat systems before use.

13. Drain all liquid tanks if dozer is to be left outside at night.

14. Check bolt torque periodically at all structural points.

15. Inspect visually for meteorite damage.

16. Check pH of electrolyte periodically.

17. Clean dust off loader every "day".

Recommendations:

1. Operator should have a moon buggy for transpeed.
2. Chain should be adjusted periodically.
3. Do not fill fuel tanks simultaneously.
4. Have radiation facility for cooling water.
5. Have a splitter for the H_2 and O_2, a compressor, and a steady supply of fuel.
6. A modular computer is needed (several spares should be on hand).

Appendix A

ALTERNATE DESIGNS

Several alternate design parameters were proposed for the design of the bulldozer. The following alternatives were proposed: a linked tracking system, wire loop wheels, batteries, a space radiator, a fiber composite chassis, power screws, direct operator access , and remote control from Earth.

The Linked Tracking System: The linked tracking system was proposed because of the weak, easily crushed surface of the moon. The major merit of the linked tracking system was its large contact area; this would reduce the crushing effect on the soil while still providing enough surface shear for the necessary bulldozing resistance. Since a tracked system is quite long compared to a tire, such a system would have been able to travel over rough terrain more readily than a wheel.

The main disadvantage of a linked tracking system was its large number of moving parts. In addition to increasing the need for maintenance (to replace failed bearings), the bearings would also require special care to keep lunar dust out of the housings. On existing earth-moving equipment, treads are hinged bare metal segments; however, such segments would be quickly cold-welded on the moon.

Wire Loop Wheels: An elastic wire loop wheel which has wide flat spokes and a spirally-connected hub and rim was also proposed as an altternative traction design. When power is applied to the loaded wheel, the wheel assumes an elliptical contact area similar to that of the tracked vehicle. This system is also easily able to deform over obstacles; i.e., these wheels travel well over rough terrain. However, the disadvantage of this tire is that it is not rigid enough to withstand the bulldozing resistance generated by our design payload and blade size.

Batteries: The use of batteries was proposed in the early design stages. The ease of maintenance (i.e., recharging the battery) and minimal repair requirements were the greatest advantages of the battery; the heavy weight of the batteries was also an additional in traction. However, batteries did not meet the power and duty requirements (of 40 horsepower and 6 hours continuous duty).

Space Radiator: A space radiator was proposed to cool the battery (see preceding section). The space radiator required that the battery be thermally isolated from the dozer to minimize the dozer's influence on the batteries). This was accomplished by means of a glass epoxy enclosure which housed an internal multi-layered insulation blanket. This insulation blanket was composed of thin radiation shields -- 25 micrometer thick polyester/polyimide films metallized with silver on one side to achieve low emittance. Spacer materials were used to separate the radiation shields and minimize shield-to-shield contact: plastic and silk netting, thin sheets of foam, and embossed plastic film. An opening in the glass enclosure permitted heat rejection from the battery radiator; to minimize absorbed solar flux, 5-mil silvered teflon was used as a radiator coating. Pin fins were attached to the radiator to provide an additional radiator area parallel to the sun vector, and the total radiator sized to maintain the batteries below 30·C. An attempt was made to modify this design for fuel cells; however, the large increase in heat dissipation made such a modification too cumbersome for our size constraints.

<u>Fiber Composite Chassis:</u> Three different fiber composites were considered as chassis materials: Dupont Kevlar K-49 structural, boron, and carbon fiber composites. All of the above materials have a strength-to-weight ration greater than that of steel; however, the epoxies used in these composites tend to fail at high operating temperatures.

<u>Power Screws:</u> Power screws were considered as an alternative to a hydraulic system. However, the relatively slow lift times of the bucket, the bulldozing resistance of such a blade and the higher heat dissipation associated with power screws made this alternative a relatively poor choice.

<u>Direct Operator Access and Remote Control from Earth:</u> Both direct access and remote control from Earth were considered as design alternatives for operating the dozer (the astronaut is still allowed the option of riding on the dozer during operation); direct operator access was considered unsafe because of the possibilities of suit puncture and vehicle tip-over. It was also difficult to design a

comfortable compartment with easily-accessible controls that met the size constraints of the shuttle. Remote control operation from Earth was considered impractical for the following reasons:

1. the time involved to develop and produce the system.
2. the lack of clear visibility of the area surrounding the dozer.
3. the inherent time delay in decision-making.
4. maintenance and repair complications.

NASA LUNAR DOZER DESIGN ENGINEERS	APPENDIX B: DECISION MATRIX	NO.	DATE 03-05-85	SCALE	INIT. J.P.

ALTERNATIVES / DESIGN CRITERIA / WEIGHTING FACTOR	EASE OF MAINTENANCE AND REPAIR	COST OF MATERIALS & MANUFACTURING	TIME TO PRODUCE	DURABILITY	RELIABILITY & FEASIBILITY	SAFETY	OVERALL SATISFACTION
	.25	.05	.05	.15	.20	.30	1.0
WHEELS	90	80	80	70	80	85	82.5
LINKED TRACKING SYS.	75	75	70	80	75	75	75.5
FUEL CELLS	75	75	75	90	85	90	83.75
BATTERIES	80	80	80	80	75	85	80.5
I.C. ENGINES	65	85	75	50	25	50	51.75
REMOTE CONTROL	85	75	75	95	95	90	89.0
DIRECT OPERATOR ACCESS	95	85	85	85	90	70	84.0
CONTROL FROM EARTH	65	70	65	80	30	80	65.0
HYDRAULICS	80	75	75	75	80	85	80.25
POWER SCREWS	70	80	65	80	70	80	74.75

NASA LUNAR DOZER DESIGN ENGINEERS	DECISION MATRIX P. 2	NO.	DATE 03-05-85	SCALE	INIT. J.P.

ALTERNATIVES / WTG. FCTR→	.25	.05	.05	.15	.20	.30	1.0
CHAIN DRIVE	80	80	90	90	80	80	82.0
BELT DRIVE	75	85	90	50	65	60	66.0
COND. HEAT TRANSFER	90	85	80	80	75	80	81.75
RAD. HEAT TRANSFER	75	70	70	75	80	80	77.0
STAINLESS STEEL CHASSIS	80	85	85	85	85	90	85.25
FIBER COMPOSITE CHASSIS	70	80	80	75	80	90	79.75

10. APPENDICES

A.1. Calculations:

Based on a 15HP motor and a 37.3KWH total demand, the design specifications of a hydrogen-oxygen fuel cell are:

Weight of fuel cell	450 lbs
Volume of fuel cell	5.6 ft^3
Hydrogen required, per KWH at STP	18 ft^3
Oxygen required, per KWH at STP	9.0 ft^3
Volume of H_2 and O_2 at 3000psi	5.0 ft^3
Weight of O_2 and H_2 and fuel tanks	250 lbm
Total weight of fuel cell and fuel system	700 lbm

In designing a fuel cell for two 20HP -15KW motors and a 240KWH total demand, a linear extrapolation of the existing data was used to calculate the projected size and weight of the new design.

Weight of fuel cell

{450 lbs/11.2KW} * 40KW = 1607.2 lbs

Volume of fuel cell

{5.6ft^3/11.2KW} * 40KW = 20 ft^3

Hydrogen required at STP

{18 ft^3/KWH} * 6KWH = 108 ft^3

Oxygen required at STP

{9 ft^3/KWH} * 6KWH = 54 ft^3

Volume of H_2 and O_2 at 3000psi

{5 ft^3/37.3KWH} * 240KWH = 32.17 ft^3

Weight of gases and resin fiber glass tank

{250 lbs/3.3 hrs} * 6 hrs = 450lbs

Total weight, fuel cell and fuel system

1607.2 + 450 ~ 2050 lbs

Weight of fuel cell	1600 lbs
Volume of fuel cell	20 ft^3
Hydrogen required,per KWH at STP	108 ft^3
Oxygen required, per KWH at STP	54 ft^3
Volume of H_2 and O_2 at 3000psi	32.2 ft^3
Weight of O_2 and H_2 and fuel tanks	450.0 lbs
Total weight of fuel cell and fuel system	2050 lbs

EQUATIONS USED IN TRACTION CALCULATIONS:

WEIGHT/WHEEL, $W = \dfrac{z^{\frac{2n+1}{2}}(3-n)(k_c + bk_\phi)\sqrt{D}}{3}$

MAX. SAFE WEIGHT/WHEEL, $W_s = A_c\left[cN_c + \gamma z N_Q + \tfrac{1}{2}\gamma b N_\gamma\right]$

WHEEL THRUST (per wheel), $H = blc\left[1 + 2h/b\right] + W\tan\phi\left[1 + 0.64\{(h/b)\cot^{-1}(h/b)\}\right]$

COMPACTION RESISTANCE/WHEEL, $R_c = \dfrac{1}{(3-n)^{\frac{2n+2}{2n+1}}(n+1)(k_c+bk\phi)^{\frac{1}{2n+1}}}\left[\dfrac{3W}{\sqrt{D}}\right]^{\frac{2n+2}{2n+1}}$

BULLDOZING RESISTANCE/WHEEL, $R_b = \dfrac{b\sin(\alpha+\phi)}{2\sin\alpha\cos\phi}\left[2zck_c + \gamma z^2 K_\gamma\right] + \dfrac{\pi t^3\gamma(90-\phi)}{540}$

$$+ \frac{c\pi t^2}{180} + ct^2\tan(45^\circ + \phi/2)$$

where $K_c = (N_c - \tan\phi)\cos^2\phi$

$K_\gamma = \left[\dfrac{2N_\gamma}{\tan\phi} + 1\right]\cos^2\phi$

$t = z\tan^2[45 - \phi/2]$

$\alpha = \cos^{-1}(1 - 2z/D)$

SYMBOLS USED:

$n \equiv$ SOIL CONSTANT $= 1$

$c \equiv$ COEF. OF COHESION $= 0.0247$ psi

$\phi \equiv$ ANGLE OF FRICTION $= 35^\circ$

$\gamma \equiv$ UNIT SOIL WEIGHT $= 0.00836$ lb_f/in^3

SOIL DEF. CONSTANTS, $k_c = 0.5026$ p[illegible]
$k_\phi = 2.98\ \dfrac{lb_f}{in^3}$

ϕ DEPENDENT CONSTS.,

$N_c = 55$
$N_Q = 35$
$N_\gamma = 35$

b = TIRE WIDTH, D = TIRE DIAMETER, z = SINKAGE, A_c = CONTACT AREA

CALCULATIONS for Wheel Parameters:

USING AN HP41 PROGRAM, "WHEEL (downloaded 3/85), THE FOLLOWING RESULTS WERE OBTAINED:

for $D = 36''$, $z = 3.25''$, $b = 12''$

$W = 851$ lbf/wheel
($A_c = 194.5\ in^2$)
$W_s = 791$ lbf/wheel ← no good; $W > W_s$

⋮

$D = 36''$, $z = 3''$, $b = 12''$

$W = 755$ lbf/wheel
($A_c = 187.6\ in^2$)
$W_s = 749$ lbf/wheel ← no good; $W > W_s$

⋮

$D = 36''$, $z = 3.25''$, $b = 14''$

$W = 991$ lbf/wheel
($A_c = 226.9\ in^2$)
$W_s = 989$ lbf/wheel ← no good; $W > W_s$

⋮

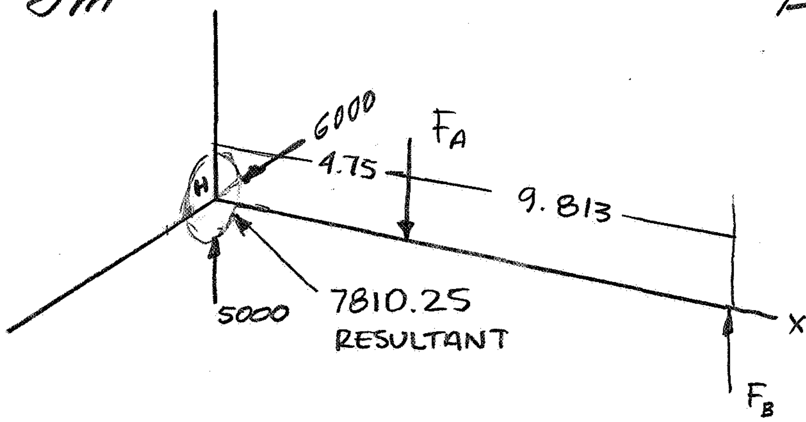
$D = 36''$, $z = 3''$, $b = 14''$

$W = 879$ lbf/wheel
($A_c = 218.8\ in^2$)
$W_s = 937.4$ lb/wheel ← ok!

$$\left.\begin{array}{l} R_c = 190 \text{ lbf/wheel} \\ H = 665 \text{ lbf/wheel} \\ R_b = 151 \text{ lbf/wheel} \end{array}\right\} DP \equiv \text{net bulldozing resistance} = 324 \text{ lbf/wheel}$$

Jm '85

AXLE



$\Sigma M_A = 4.75(7810) - 9.813(F_B)$

$F_B = \frac{4.75(7810)}{9.813} = 3{,}780 \text{ lb}$

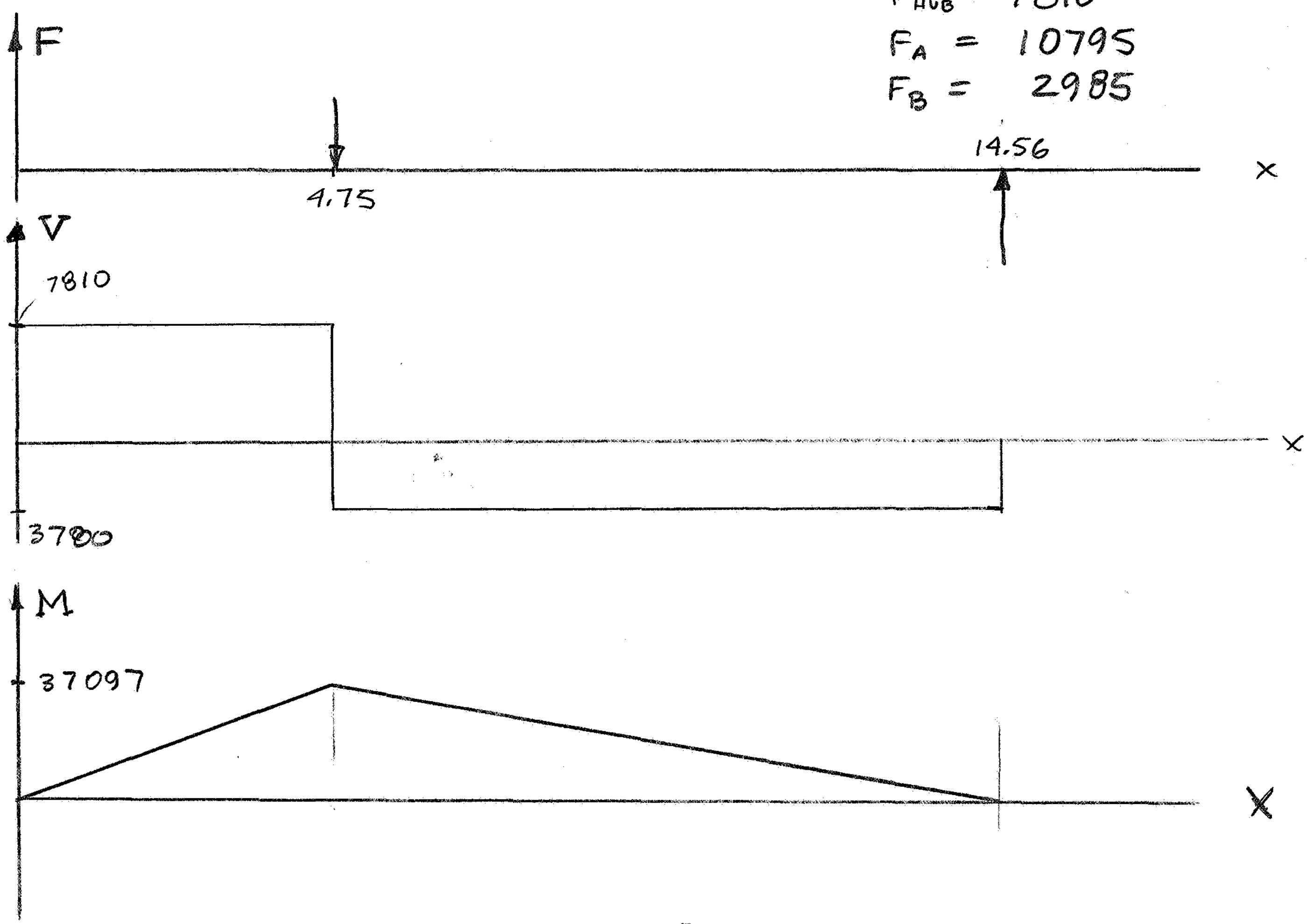
$\Sigma F = 0 = 7810 + F_A + 3780$

$F_A = -11{,}590 \text{ lb}$

$F_{HUB} = 7810$

$F_A = 10795$

$F_B = 2985$



AISI 1045 Q & T At 500°F

$S_y = 245$ kpsi

$S_u = 265$ kpsi

$S_e' = 0.5(265) = 132.5$ kpsi

$K_a = 0.89$ for Ground

$k_b = 0.869(d)^{-0.097} = 0.869(2.50)^{-0.097} = 0.795$

$K_c = 0.868$

$K_e = \frac{1}{1 + 0.92(K_t - 1)} = 0.420$

$K_t = 0.92, r = 0.12$ $\quad K_{t\,max} = 2.5$

$$S_e = 0.89\,(0.795)(0.868)(1)\,(0.420)\,132.5$$
$$= 34.178 \text{ kpsi}$$

$$\tau_m = \frac{16\,T}{\pi\, d^3} = \frac{16\,(18000)}{\pi\,(2.50)^3} = 5867 \text{ kpsi}$$

$$\sigma_a = \frac{32\,(37097)}{\pi\,(\quad)^3} = 24.183$$

$$\sigma_m' = \left[(\sigma_a)^2 + 3\,(\tau_m)^2\right]^{1/2} = 26.231 \text{ kpsi}$$

$$S_m = \frac{34.178}{\left(\frac{24.183}{26.231}\right) + \left(\frac{34.178}{265}\right)} = 32.523$$

$$n = \frac{S_m}{\sigma_m'} = \frac{32.523}{26.231} = 1.240 \quad \underline{\underline{\text{SAFE}}}$$

APPENDIX

calculation of the area of an optical solar reflector capable of getting rid of 18KW

$Q = \varepsilon A \sigma T^4$

ε: emissivity

A: Area

Q = heat

$\sigma = 5.67 \times 10^{-8} \frac{W}{m^2 K^4}$

T: temperature

$$A = \frac{Q}{\varepsilon \sigma T^4}$$

assuming

$\varepsilon = 0.8$

$T_{max} = 100°C$

$$A = \frac{18000}{(0.8)(5.67 \times 10^{-8})(373)^4}$$

$A = 20.5\ m^2$

Calculation of the amount of water storage

18KW × 6hrs = 108 KW-h = 388,800 KJ

Assuming initial temperature of water $T = 5°C$

$T_{final} = 85°C$

$Q = m c_p \Delta T$

$$m = \frac{Q}{c_p \Delta T} = \frac{388{,}800}{4.18\ (80)} = 1162.7\ Kg_{H_2O}$$

In terms of Volume $V = 1.2\ m^3$

In terms of US gallons $V = 317$ gallons

APPENDIX

The program using the HP41C to calculate the area (using iterations) is as follows.

```
LBL^T QT
Ø
STO 15
AREA = ?
PROMPT
STO Ø1
Tc = ?
PROMPT
STO Ø5
Q INIT = ?
PROMPT
STO Ø3
LBL Ø1
RCL Ø5
273
+
ENTER↑
4
Y↑X
5.67 E-8
*
.8
*
RCL Ø1
*
10000
/
STO Ø8
CHS
RCL Ø3
+
STO Ø3
1162.7
/
4.18
/
3600
*
STO Ø4
5
+
STO Ø5
.1
ST+ 15
RCL 15
FIX 1
^T HR =
ARCL X
AVIEW
FIX 2
RCL Ø5
T NEW =
ARCL X
AVIEW
RCL 15
ENTER↑
18
X>Y?
GTO 01
RCL Ø5
^T T FINAL =
ARCL X
AVIEW
END
```

HR	$A = 5 m^2$	TEMP	$A = 10 m^2$	TEMP	$A = 15 m^2$	TEMP
0		85		85		85
1		82.2		79.5		76.7
2		79.6		74.3		69.1
3		76.9		69.4		62.3
4		74.4		64.8		55.9
5		72.0		60.4		50.0
6		69.6		56.3		44.5
7		67.3		52.3		39.4
8		65.1		48.6		34.6
9	AREA = $5 m^2$	62.9	AREA = $10 m^2$	44.9	AREA = $15 m^2$	30.1
10		60.7		41.5		25.8
11		58.6		38.2		21.8
12		56.6		35.1		18.0
13		54.6		32.1		14.4
14		52.7		29.2		11.0
15		50.8		26.4		7.7
16		49.0		23.7		4.6
17		47.1		21.0		1.6
18		45.4		18.5		-1.3

Spur Gears

Using program "FW+"

(a)

Inputs:

Horsepower = 20

RPM = 70

teeth = 45

$Y = .46774$ [$\phi = 25°$, full-depth]

$S_p = 30000$

Safety factor = 3 to 4 for many materials

Outputs:

$P = 6$

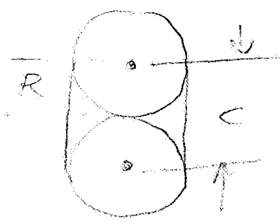
$FW = 2.288$

$FW, min = 1.571$

$FW, max = 2.618$

$$\boxed{\begin{array}{l}\text{Spur Gears} \\ N = 45 \\ \phi = 25°\text{, full-depth} \\ P = 6 \\ FW = 2.25''\end{array}}$$

Silent Chain



chain length = $2C + \pi D$

$R = \frac{D}{2} + \frac{1}{P}$

$R = \frac{N}{2P} + \frac{1}{P} = \frac{N+2}{2P}$

$C > 2R = \frac{N+2}{P}$

$C > \frac{N+2}{P}$

$P_c = \frac{\pi}{P} \rightarrow P = \frac{\pi}{P_c}$

Deriving Equations

$C > \frac{N+2}{\frac{\pi}{P_c}}$

$\boxed{N < \frac{C\pi}{P_c} - 2}$ Given C, P_c

$\boxed{D = \frac{N P_c}{\pi}}$

$\boxed{R = \frac{(N+2)P_c}{\pi}}$ Given N, P_c

Use $\boxed{P_c = 0.75''}$

$C < 7.5''$

$\rightarrow N < \frac{7.5\pi}{0.75} - 2$

$N < 29.41$

$\rightarrow D < 7.02$

Use $N = 28$, then $R = 3.58$

$\rightarrow D = \frac{28(.75)}{\pi} = \frac{21}{\pi}$

Length = $2C + \pi D$

$= 2(7.5) + \pi\left(\frac{21}{\pi}\right) = 36''$

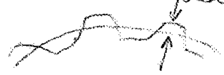
$$\boxed{\begin{array}{l}\text{SILENT CHAIN} \\ \text{Pitch} = 0.750'' \\ \text{Length} = 36'' \\ \text{Width} = 2.25'' \\ N_s = 28\end{array}}$$

NO "SCHAIN" RUN. STRENGTH ASSUMPTION.

ORIGINAL PAGE IS OF POOR QUALITY

Splines — Deriving Equations

addendum (measured from pitch circle)



"a" usually $= \frac{1}{P}$ (spur gears); however, for splines teeth are generated such that $a = \frac{1}{2P}$.

$$\rightarrow OD = D + 2\left(\frac{1}{2P}\right) = D + \frac{1}{P}$$

$$OD = \frac{N}{P} + \frac{1}{P} = \frac{N+1}{P}$$

$$\boxed{OD = \frac{N+1}{P}} \quad \boxed{N = (OD)(P) - 1}$$

$$\boxed{P = \frac{N+1}{OD}}$$

Spline rating (from GEAR DRIVE SYSTEMS)

$$\tau_{shear} = \frac{8\ \text{Torque}}{D^2 \pi\ FW}$$

Allowable shear stresses are 50000 psi for case hardened steel splines (R_c 60) and 40000 psi for R_c 33 to 38 splines.

$$\sigma_{compressive} = \frac{2(\text{Torque})\,K_m K_a}{9\,D^2\,FW\,L_f}$$

K_m — load distribution factor

K_a — application factor

L_f — fatigue-life factor

Allowable compressive stresses are 20000 psi for case hardened steel splines (R_c 60) and 12000 psi for R_c 33 to 38 splines. These values are for splines with crowned teeth.

(c)

Spline Load Dist. Factor (Km)

Misalignment (in/in)	FACE WIDTH OF SPLINE (inches) 0.5	1.0	2.0	4.0
.001	1	1	1	1.5
.002	1	1	1.5	2
.004	1	1.5	2	2.5
.008	1.5	2	2.5	3

Spline Application Factor (Ka)

Power Source	Load Type: Uniform	Light shock	Medium	Heavy
Uniform	1	1.2	1.5	1.8
Light	1.2	1.5	1.8	2.1
Medium	2	2.2	2.4	2.8

Spline fatigue-life factor (Lf)

# torque cycles	Unidirectional	Fully Reversed
10^3	1.8	1.8
10^4	1	1
10^5	.5	.4
10^6	.4	.3
10^7	.3	.2

Use case-hardened steel with R_c 60 hardness cd.

Splines specified from diagrams (stress check)

Input Shaft

Input lock spline:

$N = 14$
$P = 10$
$OD = 1.5$
$D = 1.4$
$FW = 1.5''$

$$\sigma_s = \frac{(8)(18000)}{(1.4)^2 \pi (1.5)} = 15590 \text{ psi}$$ ✓

well below limit of 50000

$$\sigma_c = \frac{2(18000)(1)(1.5)}{9(1.4)^2(1.5)(.3)} = 6800 \text{ psi}$$ ✓ well below 20000

Main Spline:

$N = 16$
$P = 8$
$OD = 2.125''$
$D = 2''$
$FW = 2.5''$

$$\sigma_s = \frac{8(18000)}{(2)^2 \pi (2.5)} = 4580 \text{ psi}$$ ✓

$$\sigma_c = \frac{2(18000)(1)(1.5)}{9(2)^2(2.5)(.3)} = 2000 \text{ psi}$$ ✓

Output Shaft

Output Sprocket Spline:

$N = 15$
$P = 8$
$OD = 2''$
$D = 1.875''$
$FW = 3''$

$$\sigma_s = \frac{8(18000)}{(1.875)^2 \pi (3'')} = 4350 \text{ psi}$$ ✓

$$\sigma_c = \frac{2(18000)(1.5)(1.5)}{9(1.875)^2(3)(.3)} = 2840 \text{ psi}$$ ✓

Free-spinning splines on gears/sprockets:

$N = 15$
$P = 8$
$OD = 2''$
$D = 1.875''$
$FW = 1''$

$$\sigma_s = \frac{8(18000)}{(1.875)^2 \pi (1'')} = 13050 \text{ psi}$$ ✓

$$\sigma_c = \frac{2(18000)(1)(1.5)}{9(1.875)^2(1)(.3)} = 5680 \text{ psi}$$ ✓

ROLLER CHAIN   Use "CHAIN" program

Axle - axle center distance = 60.0"

Center distances from gearbox output to rear axles : 17.5" and 30.0"

Common Inputs :

HP, REQ = 20
RPM, SMALL = 70
RPM RATIO = 1
SRVC FCTR = 1
ANSI NO = 100
STRANDS = 2

SPECIFIC INPUTS :

SH CNTRS = 60 (1)
17.5 (2)
30 (3)

Outputs :

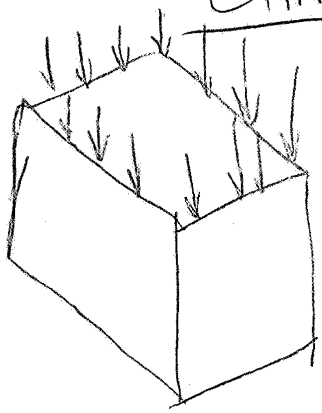
NS = 26
ANSI = 100
PITCH = 1.250
STRANDS = 2
HP, CALC = 20.11
CT, MIN = 11.04

(1) LINKS = 122
LENGTH = 152.50
CT = 60.00
} 2 of each

(2) LINKS = 54
LENGTH = 67.50
CT = 17.50

(3) LINKS = 74
LENGTH = 92.50
CT = 30.00

CHASSIS

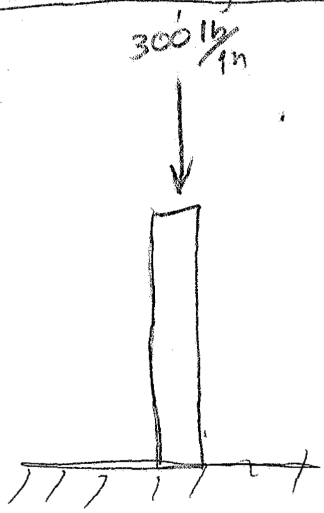


Assume uniform loading on sides

TOTAL weight 6000 lb
TOTAL AREA 20 in
stress 300 lb/in around side

Total stress $= 300\,\text{lb/in} \times \frac{1}{\frac{1}{4}\,\text{in}} = \underline{1200\ \text{psi}}$

Buckling Analysis



$$P_{CR} = \frac{\pi^2 E I}{4 \ell^2} = \frac{\pi^2 (27.6 \times 10^6\ \text{psi})}{4(36\ \text{in})^2}$$

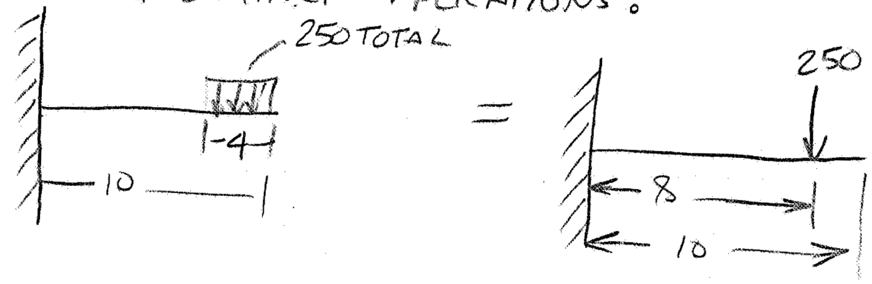
$$P_{CR} = 52.5\ \text{klbf}$$

$$\frac{52.5\ \text{klbf}}{\frac{1}{4}\ \text{in}} = \underline{210\ \text{klbf} \cdot \text{in}}$$

The chassis would easily handle the load on the moon, but they are oversized because they must withstand the total forces on earth as well. Because these panels are longer than they are tall, it is most unlikely for buckling to occurr length wise. ADDITIONALLY, The Reinforcements of 1/4" I Beams will add rigidity and added support during Liftoff in the shuttle (10 g Forces).

LIFT ARM ASSEMBLY MOUNTS

DURING LIFTING APPLICATIONS:



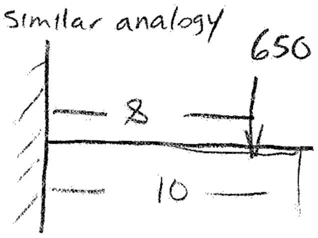
FROM SHIGLEY'S APPENDIX

$$M_1 = (-250)(8\,in) = -2000\ in\ lbf$$

$$R_1 = V = 250\ lbf$$

DURING BULLDOZING FORCES

by similar analogy



$$M = (-650\,lbf)(8\,in) = 5200\,in\,lbf$$

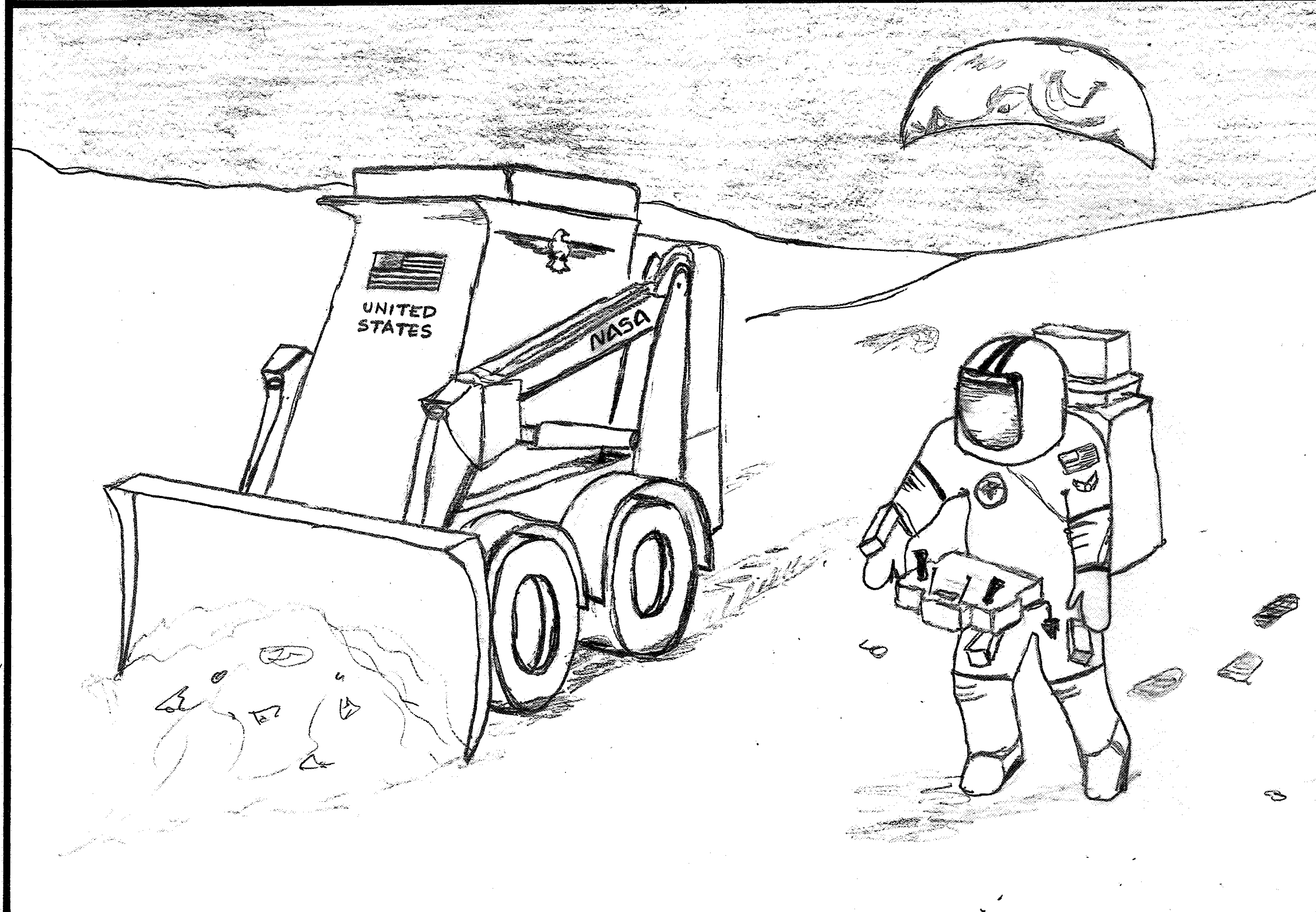
$$R_1 = V = 650\,lbf$$

Because the loads will be carried by a uniform loading at the wall the ratings are

$$\frac{\text{LOAD}}{\text{Area of X-section}} = \frac{650\,lbf}{(10\,in)(\frac{1}{8}\,in)} = \frac{650\,lbf}{\frac{10}{8}\,in^2} = \underline{520\,psi}$$

$\frac{1}{8}''$ steel will easily handle this load.

NASA LUNAR DOZER DESIGN ENGINEERS	CONCEPT DRAWING	NO. 1	DATE 3-10-85	SCALE	INIT. M.M.

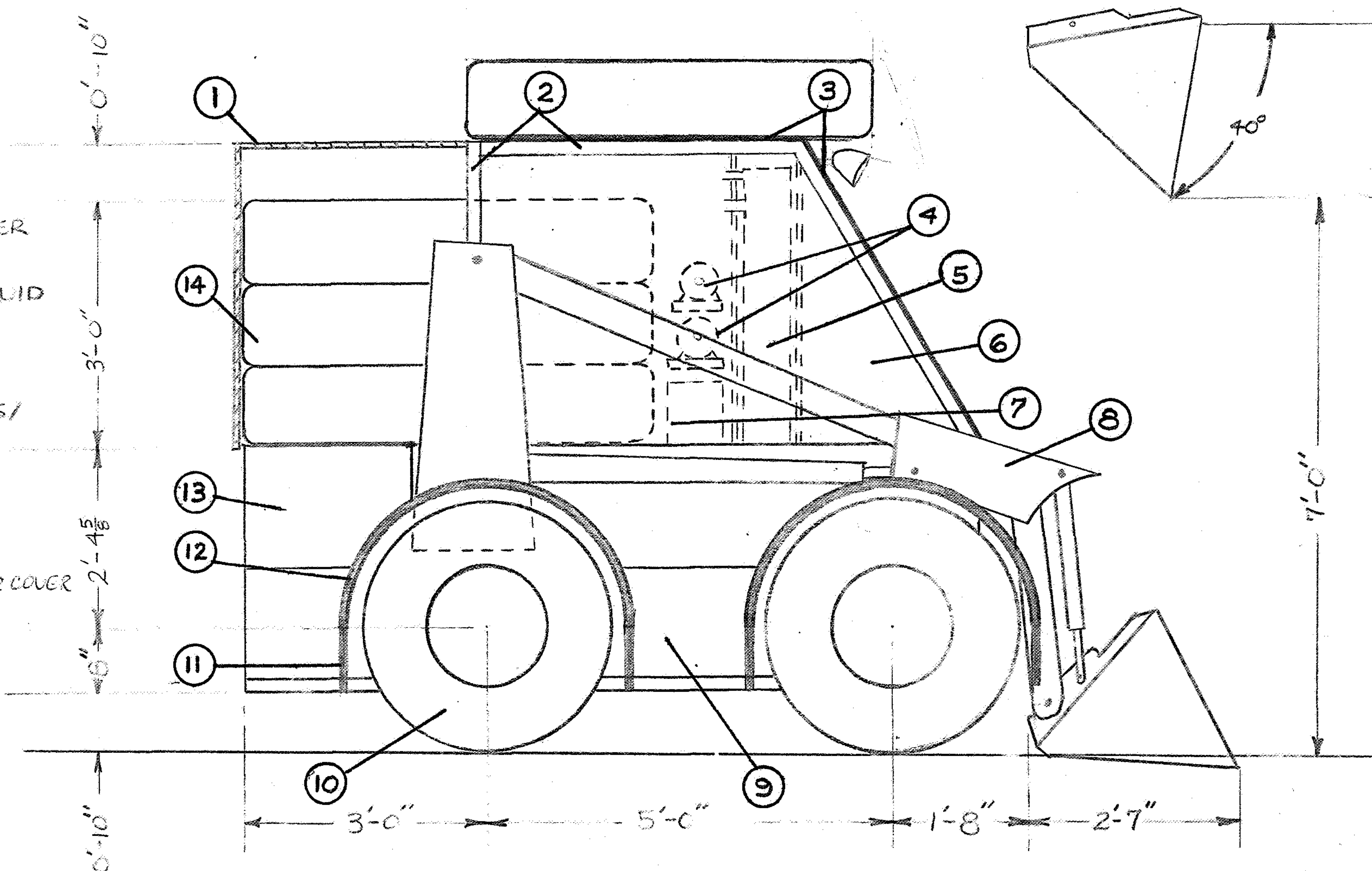


NASA LUNAR DOZER DESIGN ENGINEERS	SIDE VIEW (COMPONENTS EXPOSED)	NO. 2	DATE 05MAR85	SCALE 0'-1" = 2'-0"	INIT. N.M.

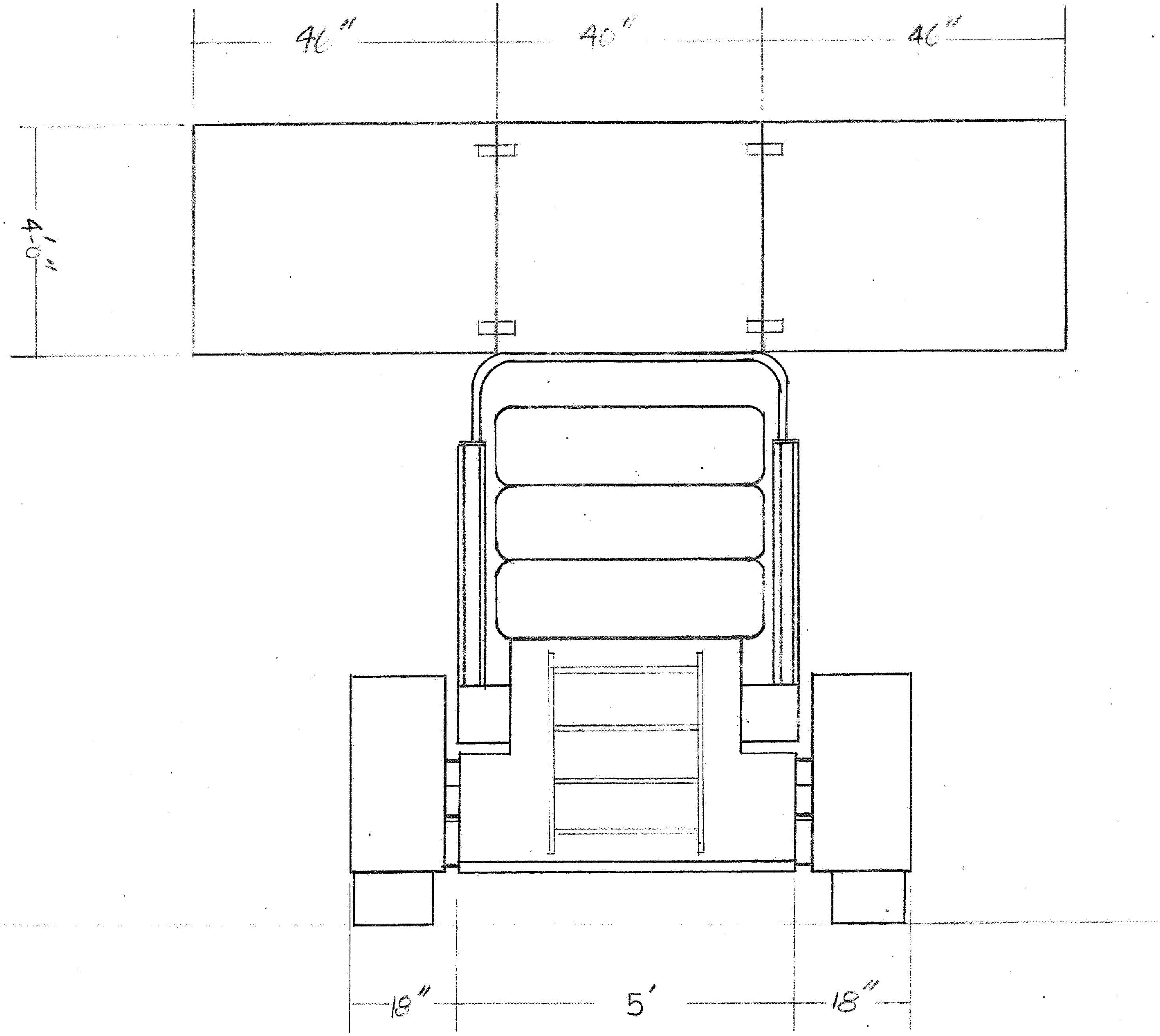
LEGEND:

- — LIFT ARM ASSEMBLY
- — SOLAR RADIATION PROTECTION
- — COOLANT RESERVOIRS
- —

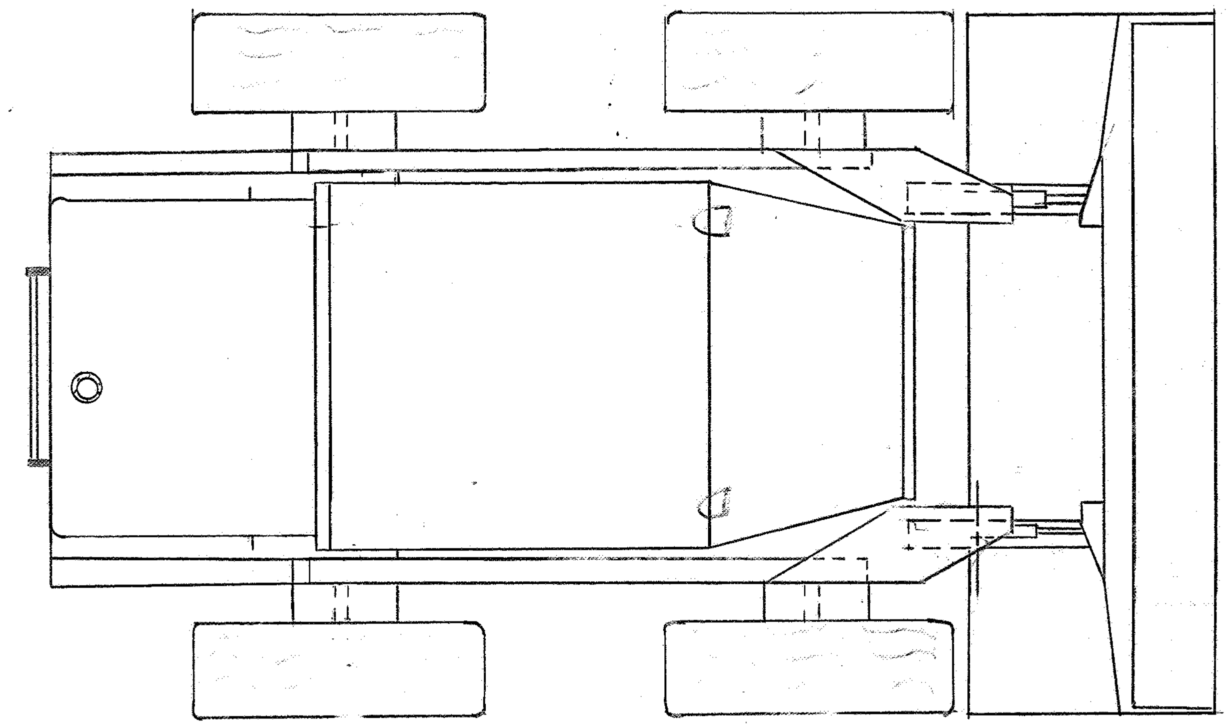
1 - SOLAR RAD. PROTECTION
2 - ROLL OVER CAGE
3 - SOLAR RAD. PROTECTION
4 - AUX PUMPS & MOTORS
5 - HEAT EXCHANGER
6 - COMPUTER
7 - HYDRAULIC FLUID RESERVOIRS
8 - LIFT ARM ASSEMBLY
9 - WHEEL MOUNTS/ DUST COVER
10 - WHEEL
11 - DUST FLAP
12 - WHEEL GUARD
13 - TRUNK/MOTOR COVER
14 - COOLANT RESERVIOR



NASA LUNAR DOZER DESIGN ENGINEERS	REAR VIEW	NO. 3	DATE 8 MAR 85	SCALE 2"=1'-0"	INIT. N.M

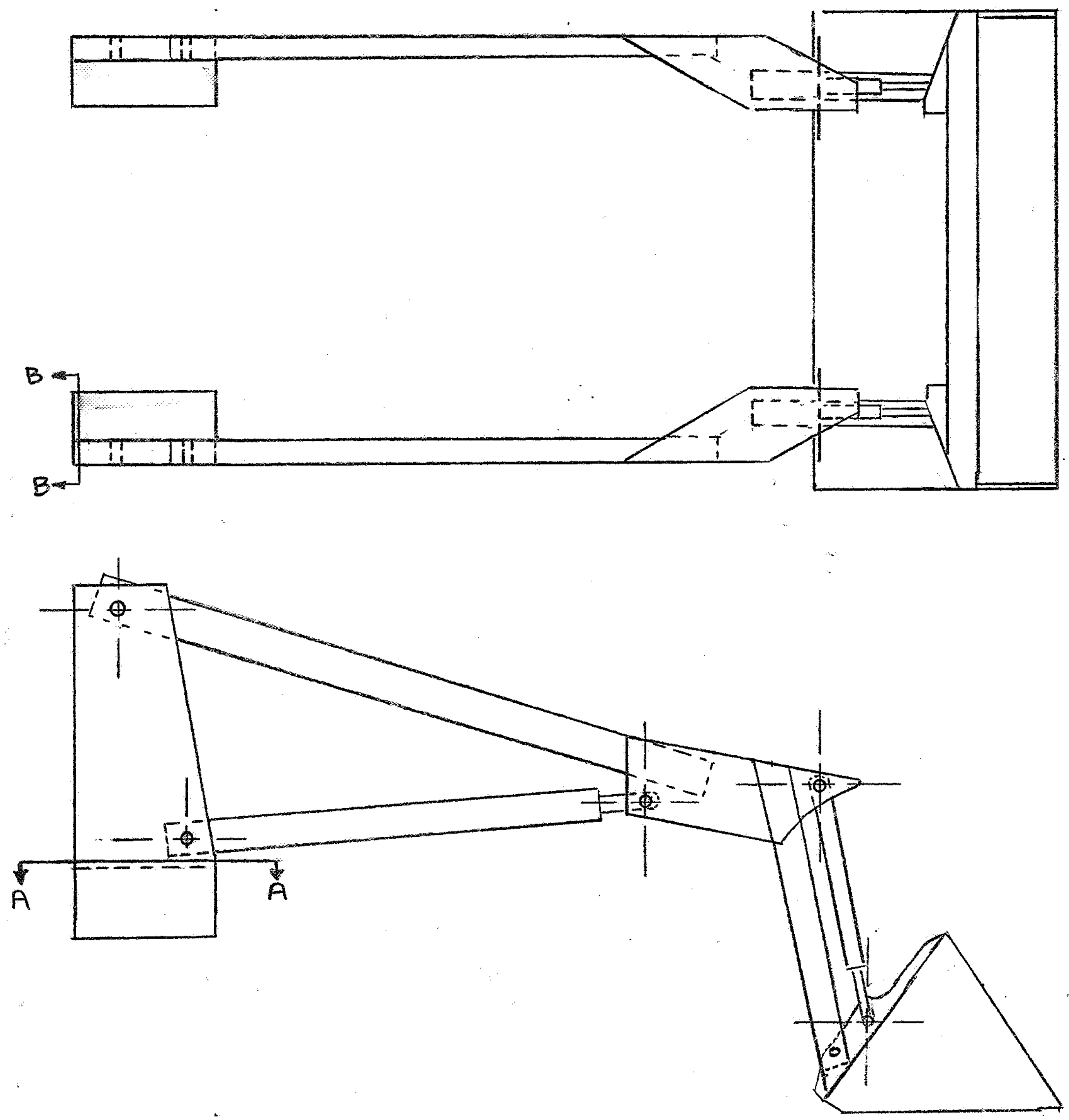


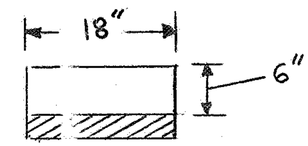
DRAWING # 4	LUNAR DOZAR (TOP VIEW)	SCALE
	DRAWN BY: N.M. DATE: 25FEB85	1" = 2'-0"



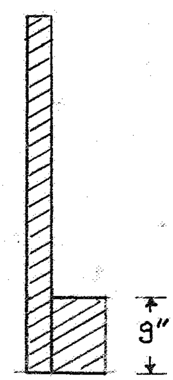
DRAWING # 5	LIFT ARM AND BUCKET	SCALE
	DRAWN BY: A-A DATE: 2124	1" = 2'-0"

THIS DESIGN, EXISTS AND IS PART OF THE CASE 1835 B UNI-LOADER. THE ONLY ADDITION IS THE WELDING OF THE TWO LOAD BEARING MEMBERS WHOSE DIMENSIONS ARE GIVEN IN SECTIONS A-A AND B-B



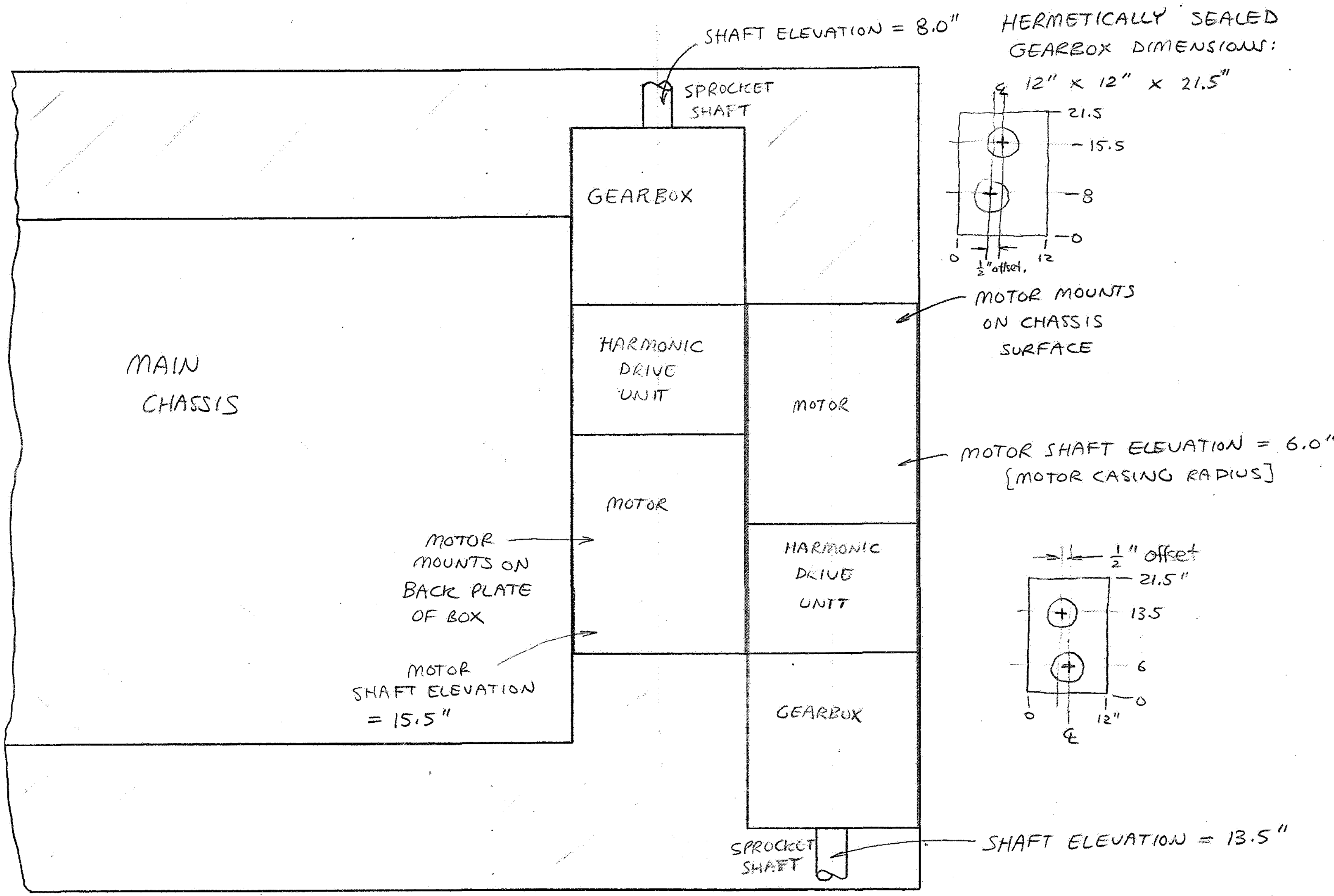


SECTION A-A



SECTION B-B.

NASA LUNAR DOZER DESIGN ENGINEERS	DRIVETRAIN MOUNTING / TOP VIEW	NO.	DATE	SCALE	INIT.
		6	04MAR85	1:10	P.T.

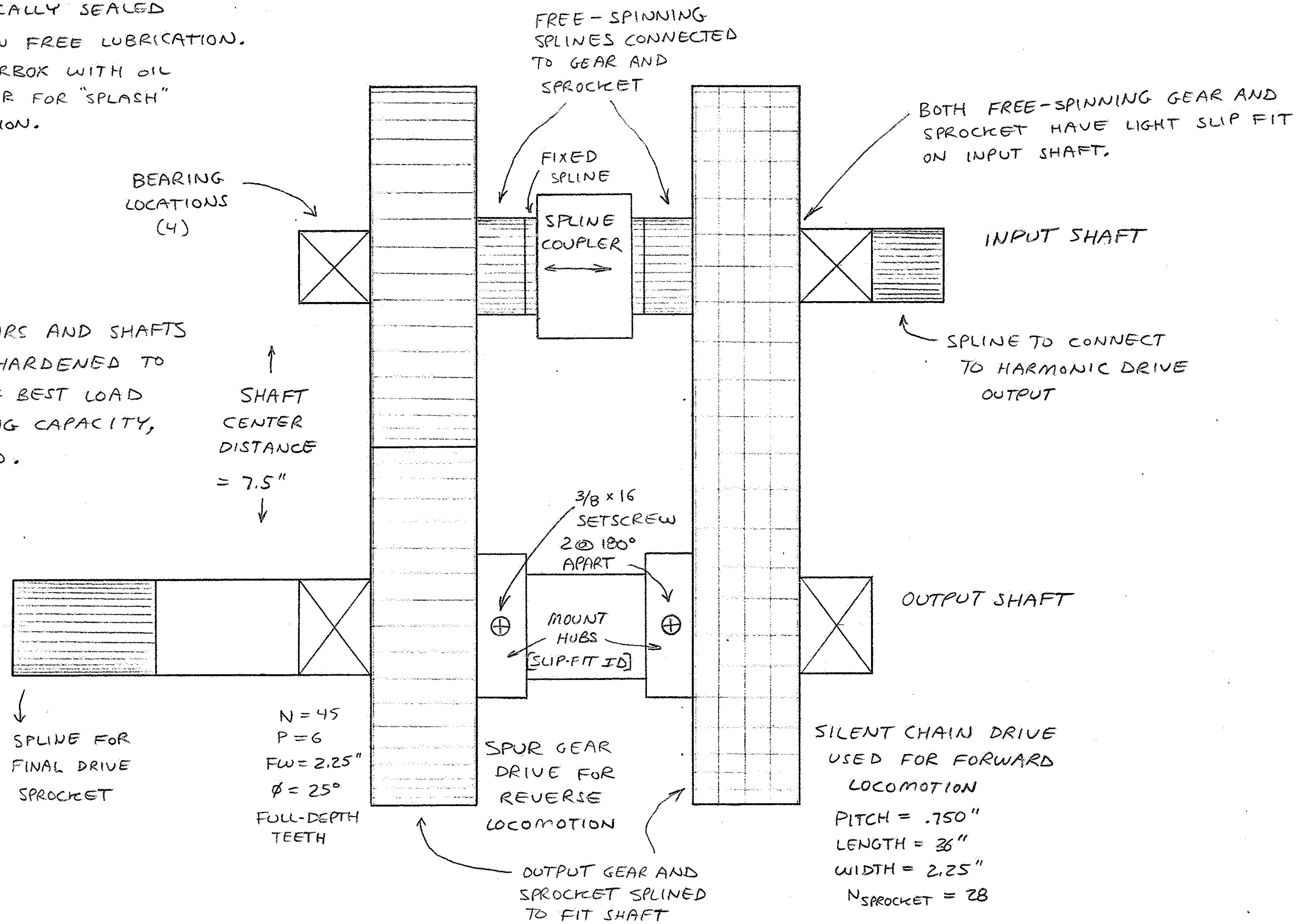


SPROCKET SHAFTS ARE OFFSET 1/2 INCH FORWARD [←] TO ACCOMODATE CENTER DISTANCE BETWEEN OUTPUT SHAFTS AND REAR AXLES.

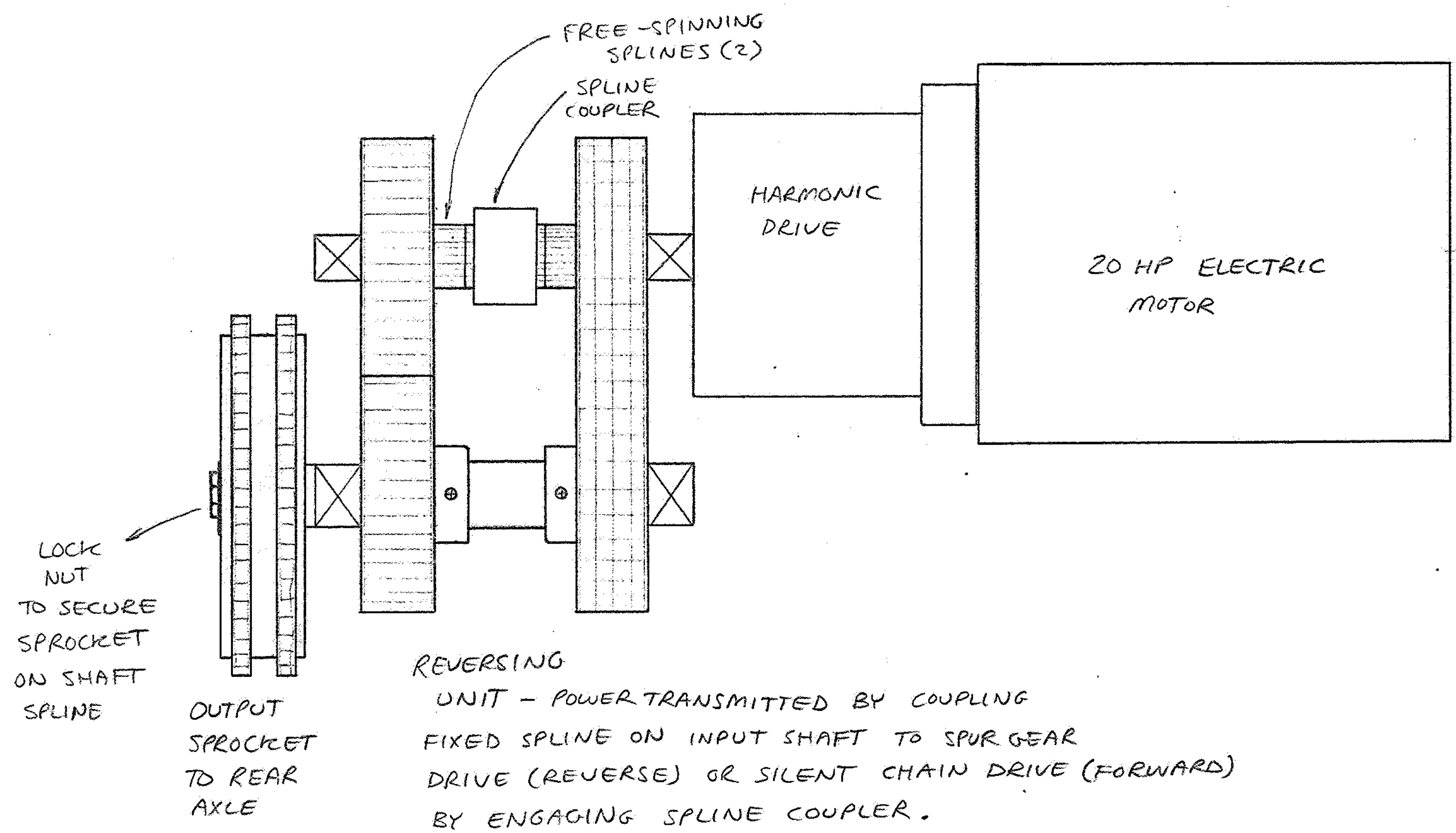
NASA LUNAR DOZER DESIGN ENGINEERS	REVERSING UNIT	NO. 6	DATE 04 MAR 85	SCALE 1:3	INIT. P.T.

GEARBOX IS TO BE HERMETICALLY SEALED TO ALLOW FREE LUBRICATION. FILL GEARBOX WITH OIL RESERVOIR FOR "SPLASH" LUBRICATION.

ALL GEARS AND SHAFTS CASE-HARDENED TO ACHIEVE BEST LOAD CARRYING CAPACITY, ~ R_c 60.

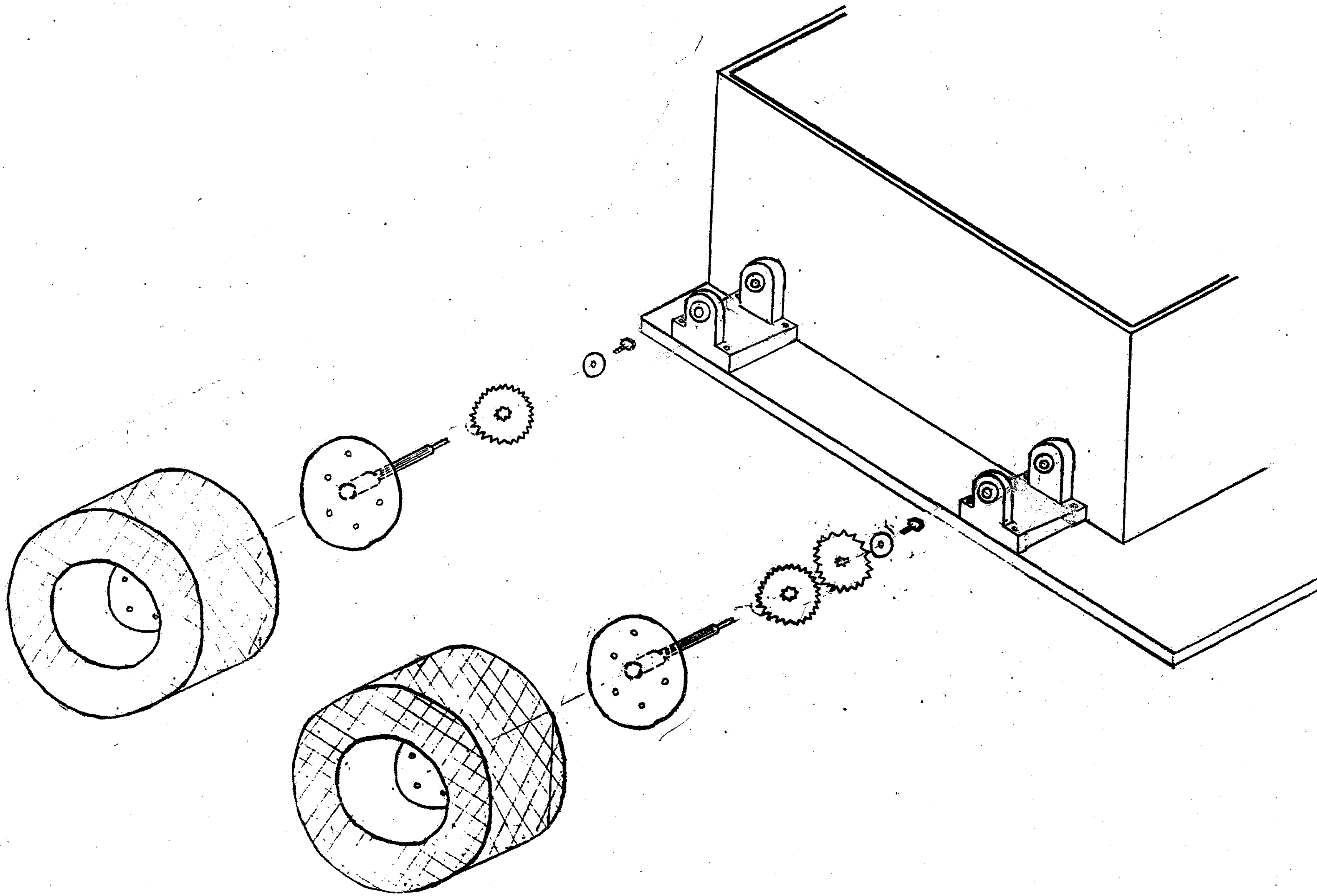


NASA LUNAR DOZER DESIGN ENGINEERS	DRIVETRAIN	NO.	DATE	SCALE	INIT.
		7	04 MAR 85	1:6	P.T.

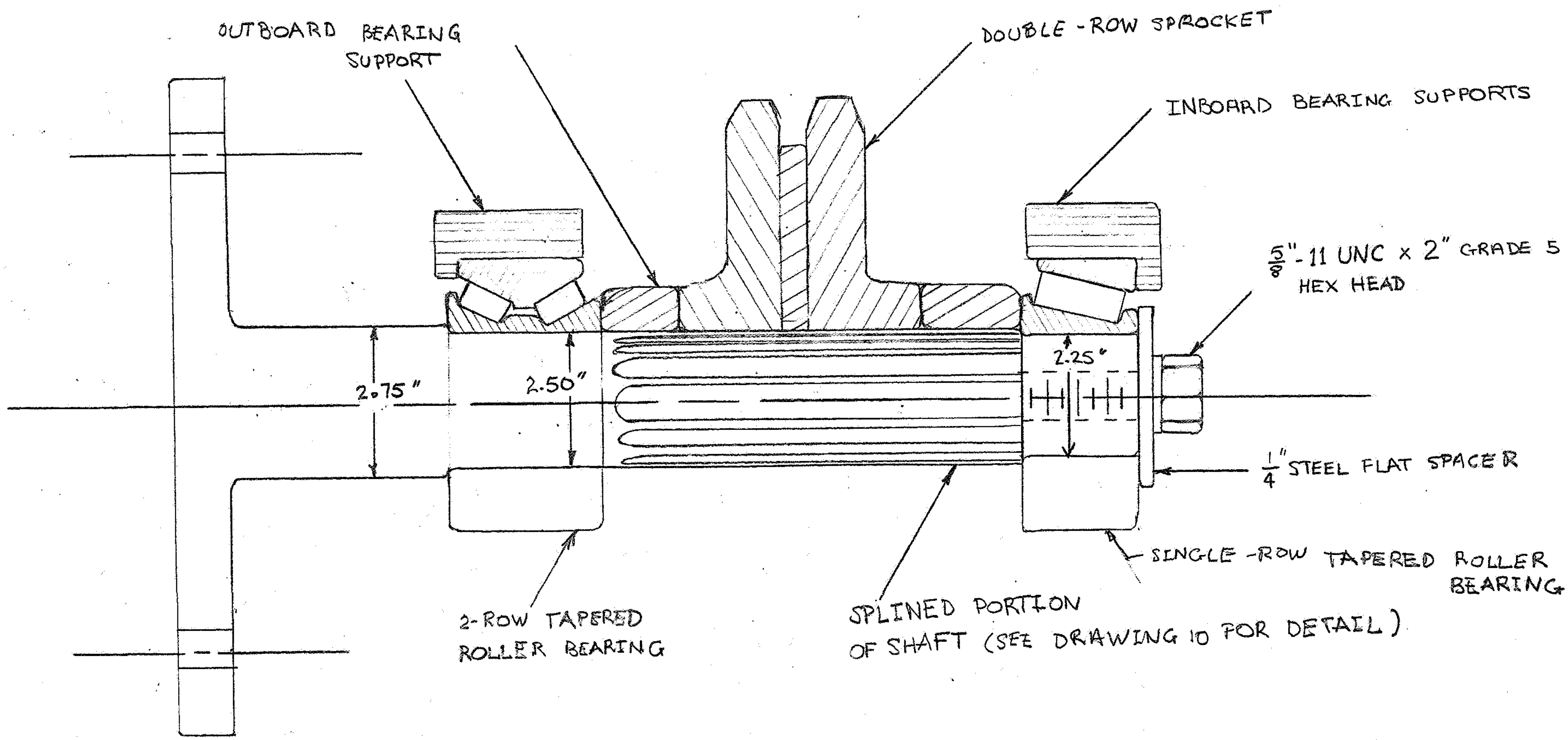


REVERSING UNIT – POWER TRANSMITTED BY COUPLING FIXED SPLINE ON INPUT SHAFT TO SPUR GEAR DRIVE (REVERSE) OR SILENT CHAIN DRIVE (FORWARD) BY ENGAGING SPLINE COUPLER.

NASA LUNAR DOZER DESIGN ENGINEERS	EXPLODED VIEW OF WHEEL ASSEMBLY	NO.	DATE	SCALE	INIT.
		8	08 MAR 85	2" = 1'-0"	N.M

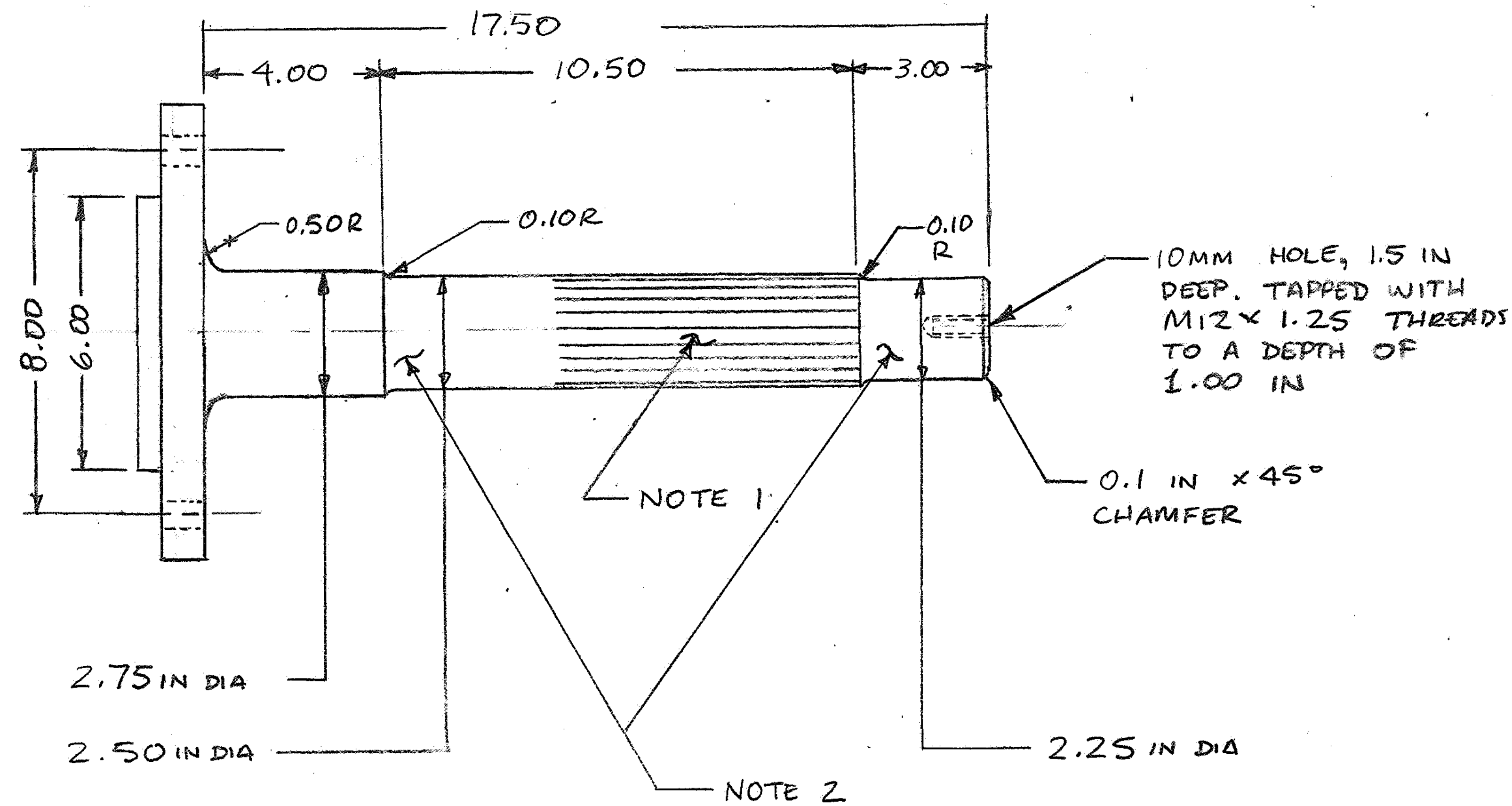


NASA LUNAR DOZER DESIGN ENGINEERS	DETAIL OF FRONT AXLE AND BEARINGS	NO.	DATE	SCALE	INIT.
		9	3-11-85	$1'' = 2\frac{1}{2}''$	C.M.T.



NOTE: LN2 FIT BETWEEN BEARING SUPPORTS AND OUTER BEARING RACES.
LC5 FIT BETWEEN SHAFT AND INNER BEARING RACES.

NASA LUNAR DOZER DESIGN ENGINEERS	AXLE	NO.	DATE	SCALE	INIT.
		10	3-6-85	1" = 4"	M.M.



NOTES:

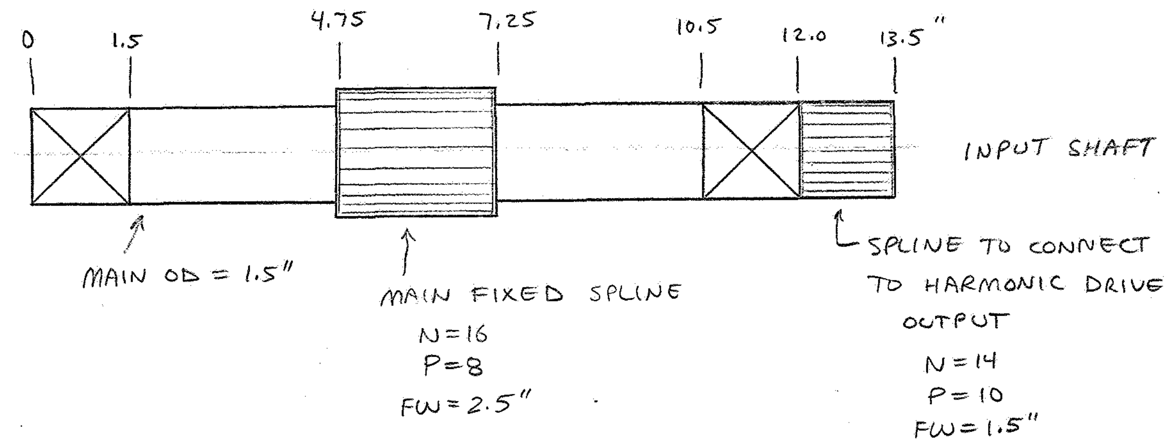
① 19 TOOTH SPLINE WITH DIAMETRAL PITCH OF 2.375 IN. TOOTH HEIGHTH OF 0.125 INCHES

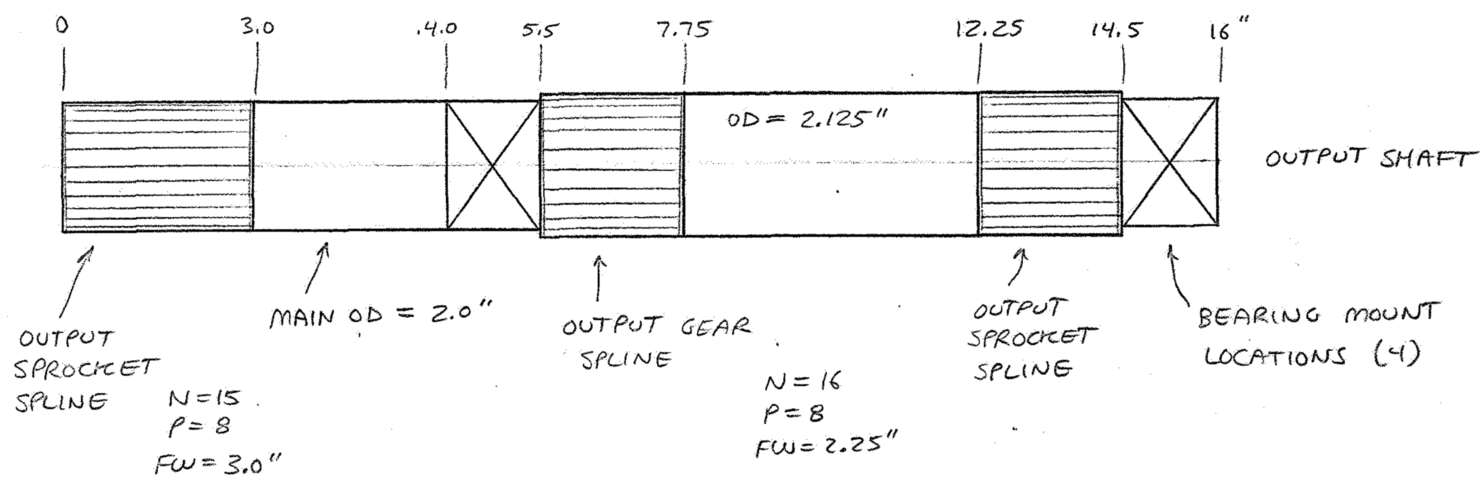
② BEARING SURFACES TO BE GROUND SMOOTH

MATERIAL: A.I.S.I. 1045 STEEL Q&T 500°F

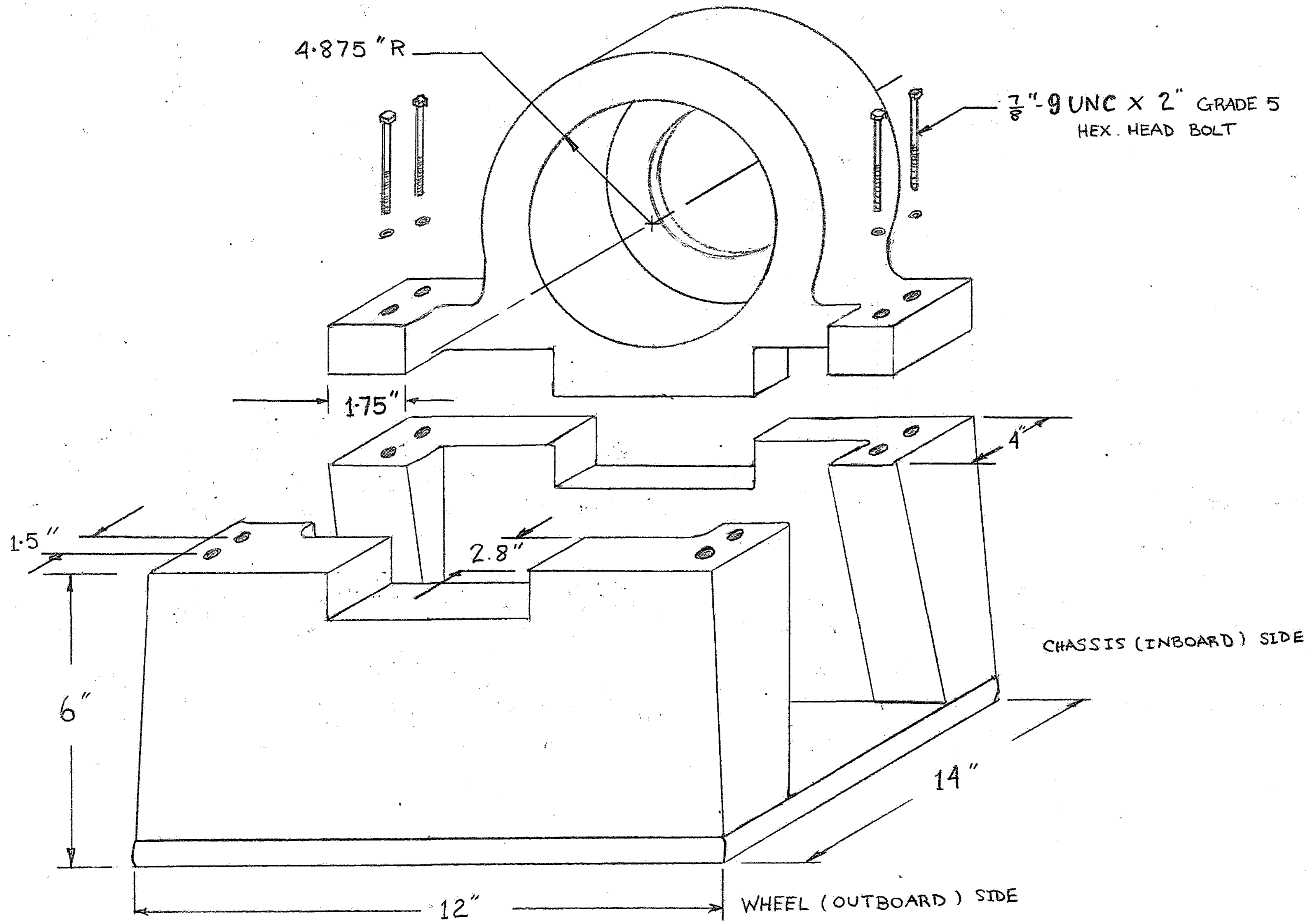
NASA LUNAR DOZER DESIGN ENGINEERS	SHAFTS IN REVERSING UNIT	NO. 11	DATE 04 MAR 85	SCALE 1:3	INIT. P.T.

ALL SHAFTS AND SPLINES
CASE HARDENED TO $\sim R_c$ 60.
ALL SPLINES CROWNED.
BEARINGS TO HAVE
WRING FIT ON SHAFTS.



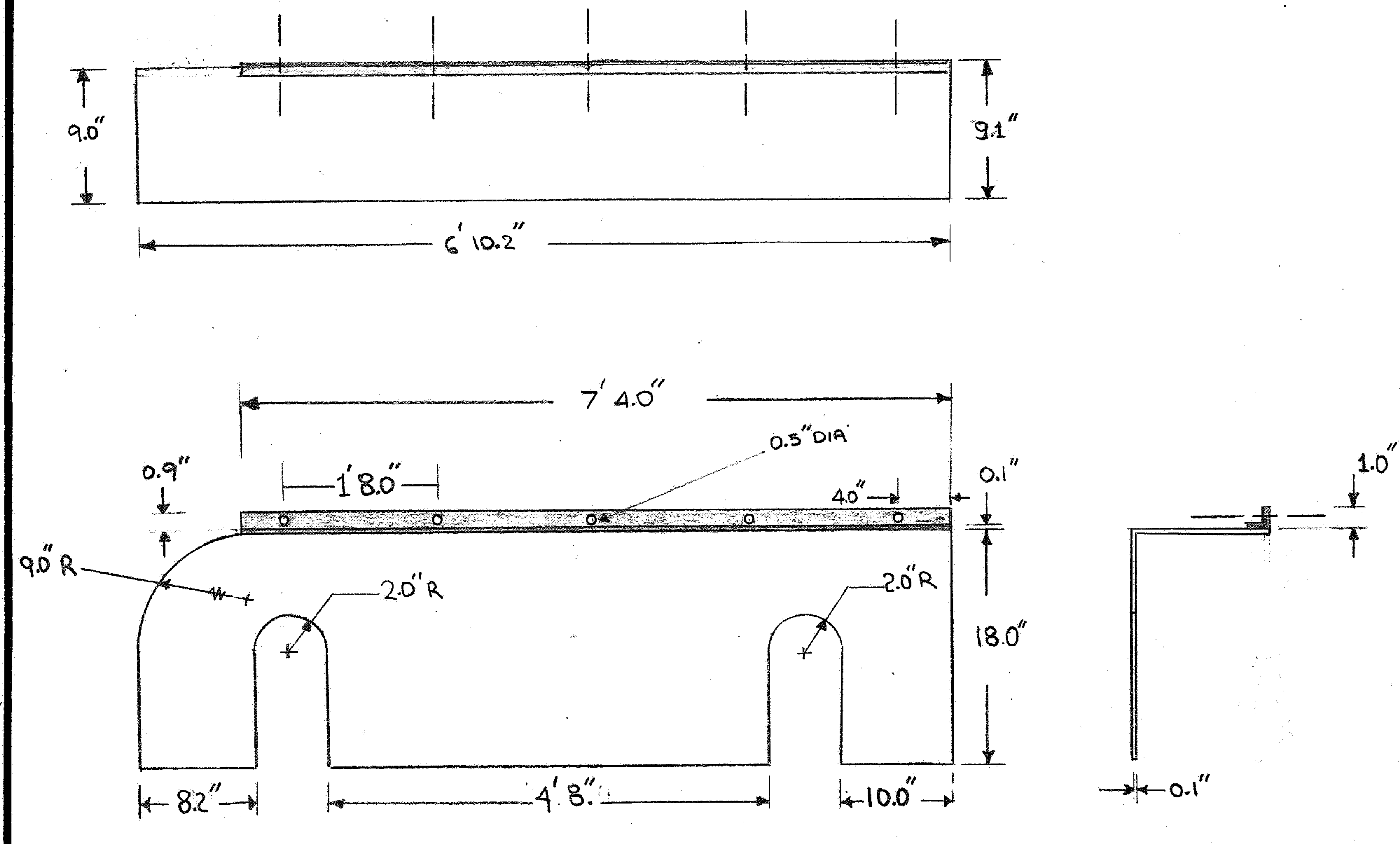


NASA LUNAR DOZER DESIGN ENGINEERS	PARTIAL EXPLODED VIEW OF AXLE MOUNT ASSEMBLY	NO. 12	DATE 3-4-85	SCALE 1:2.5	INIT. C.M.T.



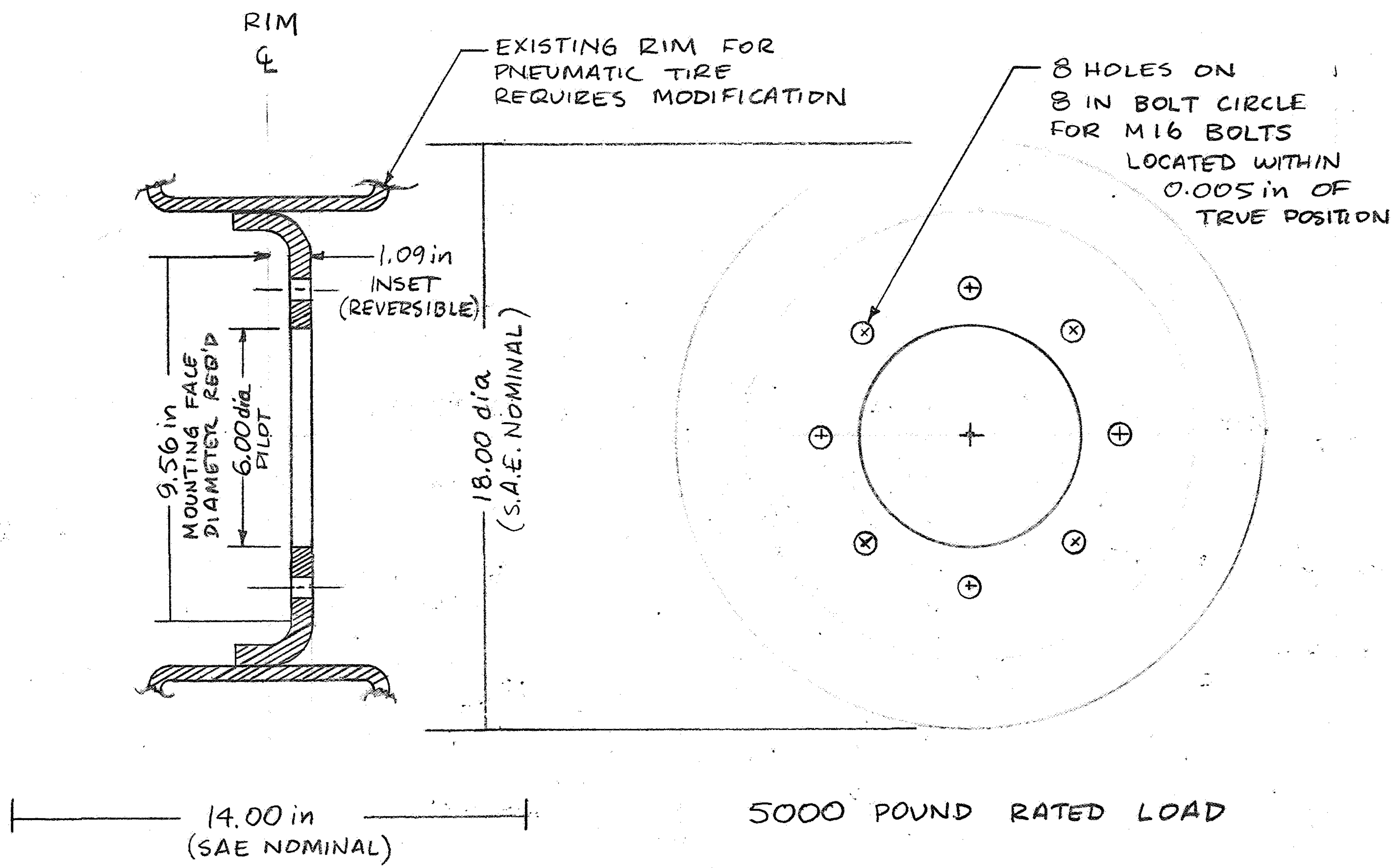
NOTE: OUTBOARD BEARING MOUNT NOT SHOWN.

NASA LUNAR DOZER DESIGN ENGINEERS	CHAIN GUARD	NO. 13	DATE 3\11\85	SCALE NOT SCALED	INIT. A.A



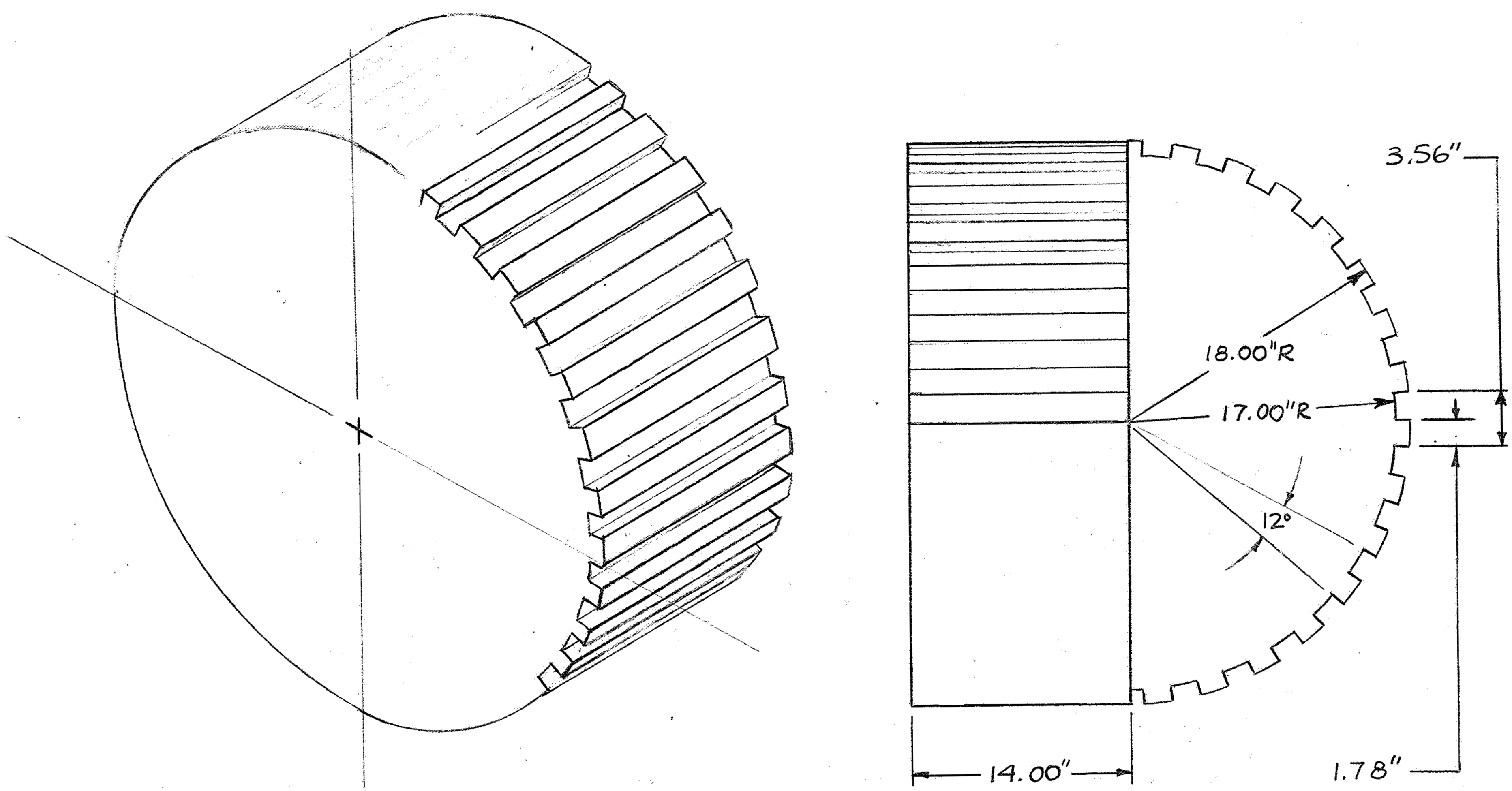
MATERIAL: A93003-O ALUMINUM 0.1" THICK PLATES

NASA LUNAR DOZER DESIGN ENGINEERS	SAE #806 DISK WHEEL SPECIFICATIONS	NO. 14	DATE 3-5-85	SCALE 1" = 4"	INIT. M.M.



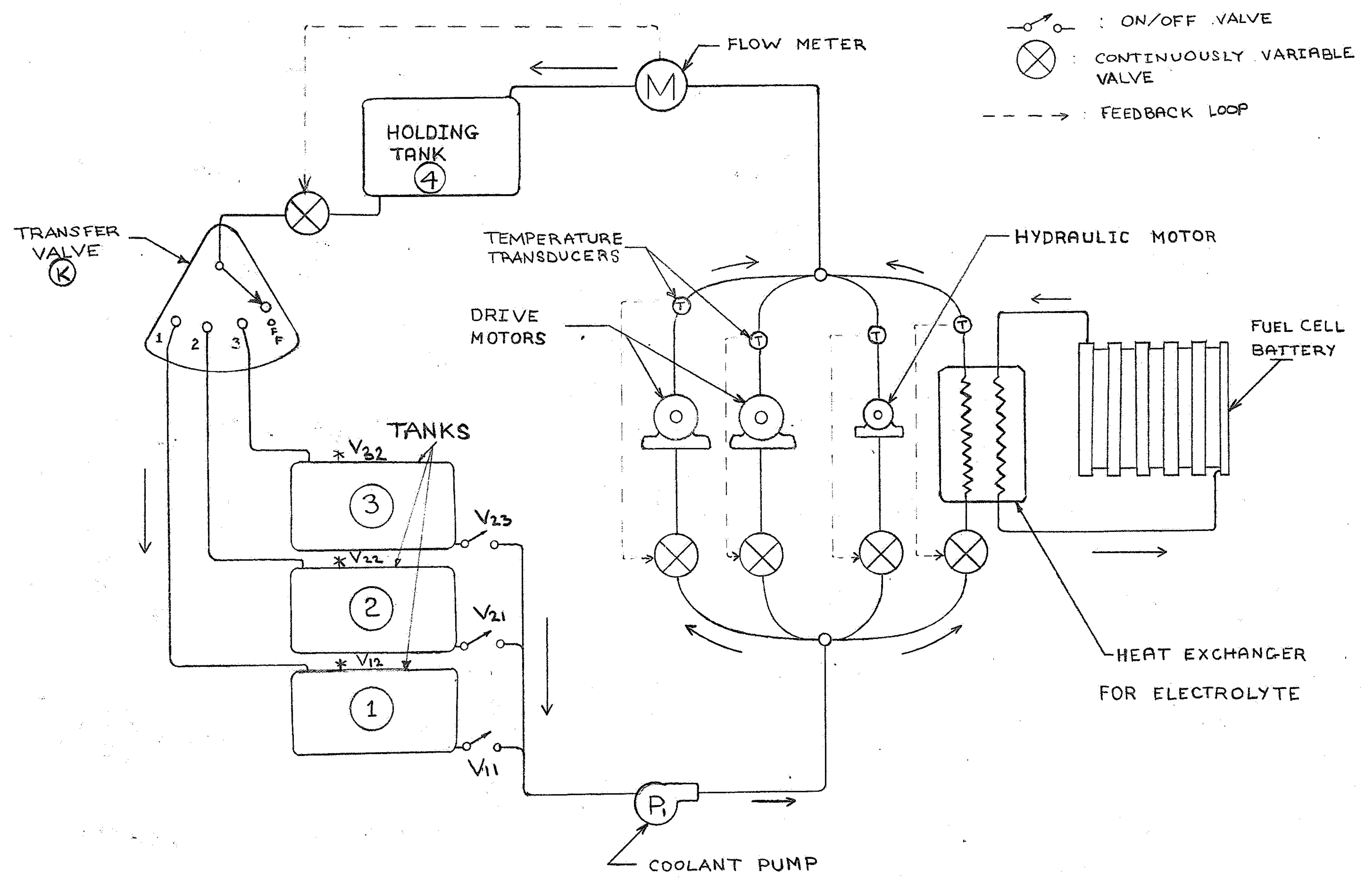
5000 POUND RATED LOAD

DRAWING #15	DRAWN BY: M.M. DATE: 2-24	SCALE 1" = 10"



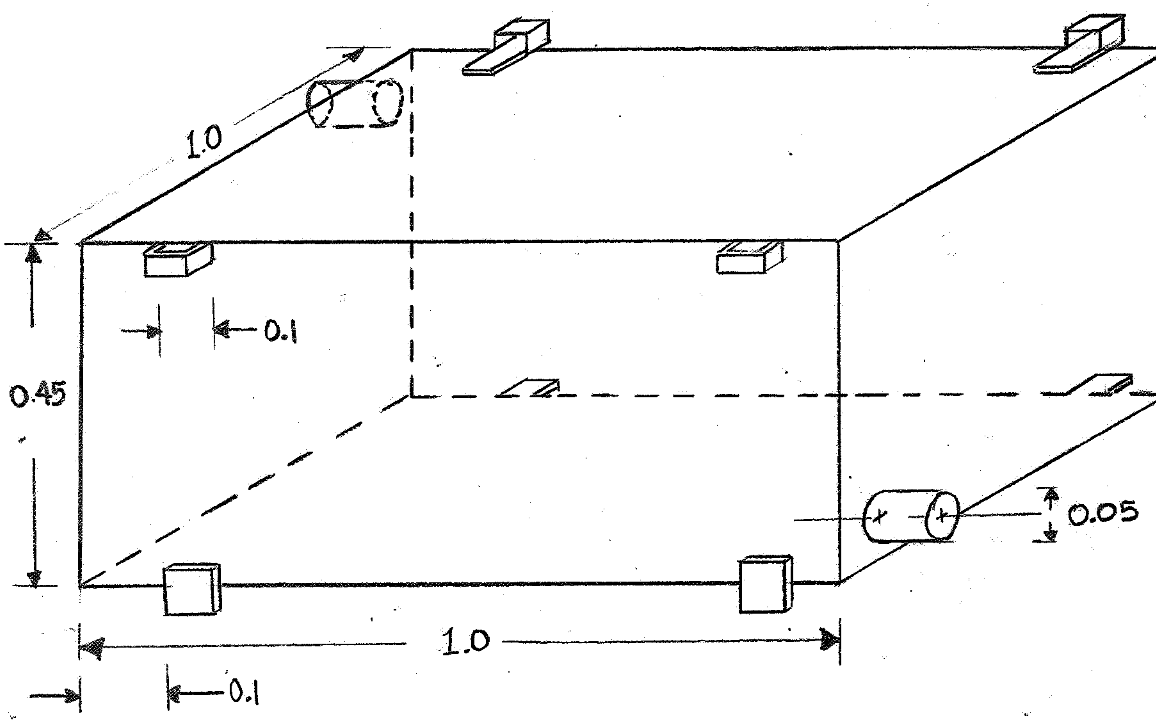
WHEEL PROFILE -
BASIS FOR
TRACTION CALCULATIONS

NASA LUNAR DOZER DESIGN ENGINEERS	COOLING SYSTEM	NO.	DATE	SCALE	INIT.
		16	3/07/85	SCHEMATIC	C.M.T.



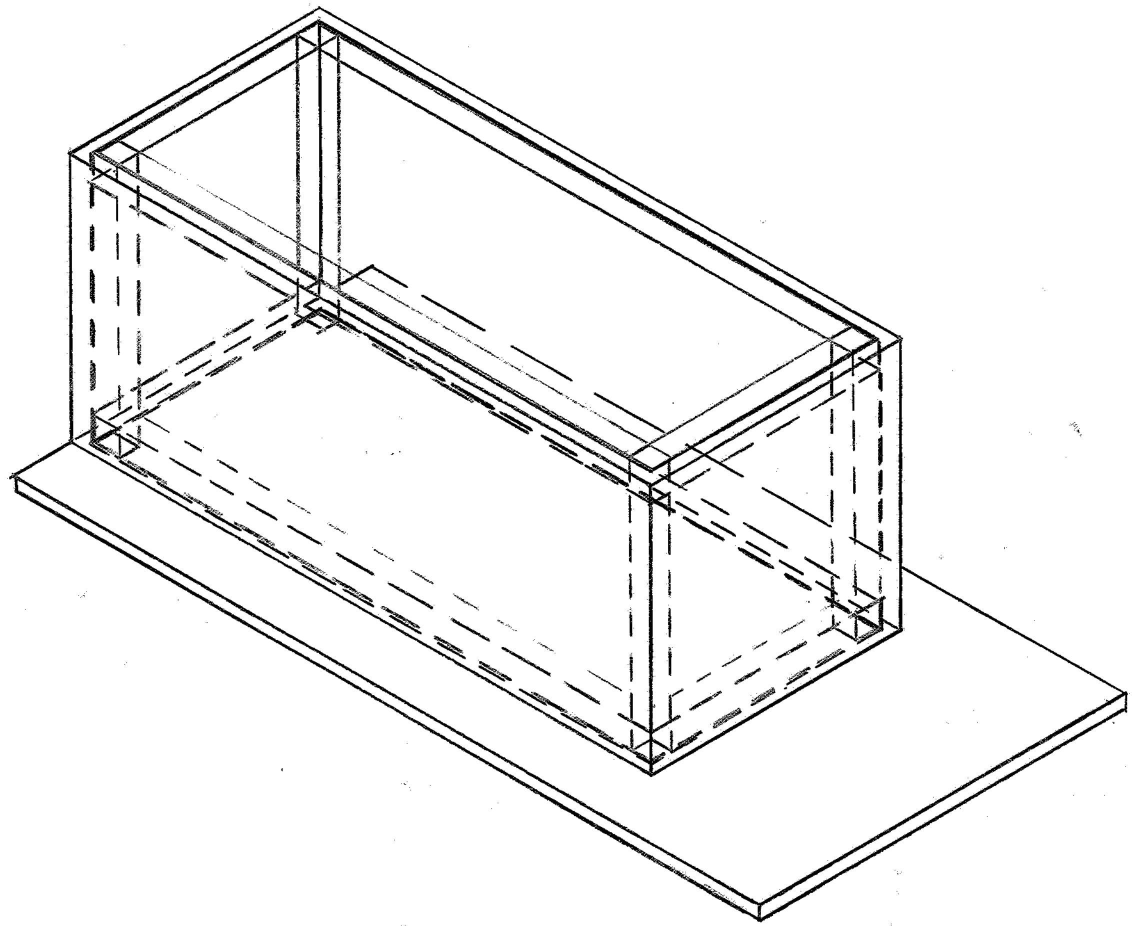
NASA LUNAR DOZER DESIGN ENGINEERS	COOLANT WATER TANKS	NO. 17	DATE 3/10	SCALE 1:10	INIT. A.A

MATERIAL : 6061-T6 ALUMINUM PLATES 3.2 mm THICK

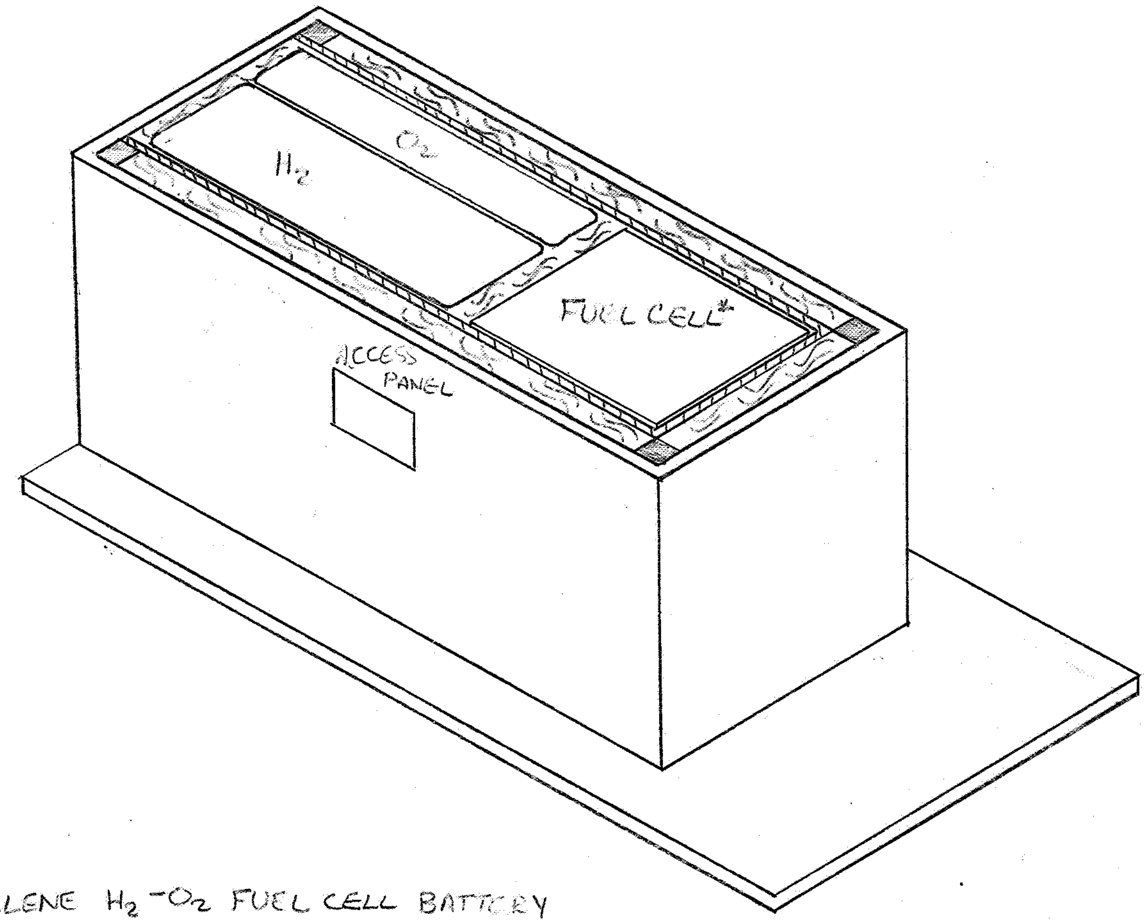


ALL DIMENSIONS IN METERS

DRAWING	CHASSIS & REINFORCEMENT	SCALE
#18	DRAWN BY: N.M. DATE: 23FEB85	1" = 2'-0"

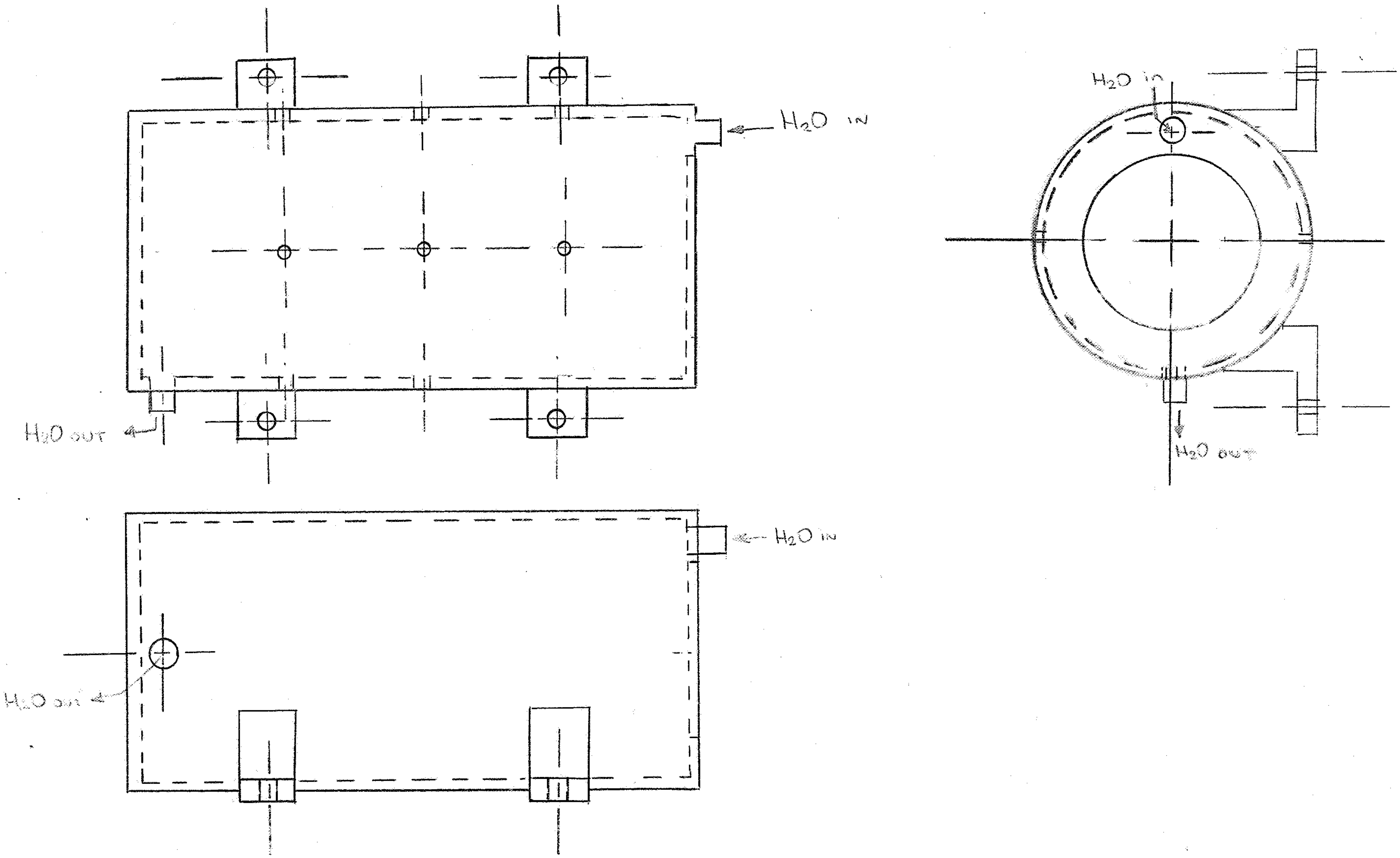


DRAWING #19	FUEL CELL & TANKS SCEMATIC	SCALE
	DRAWN BY: N.M. DATE: 23FEB85	N/A

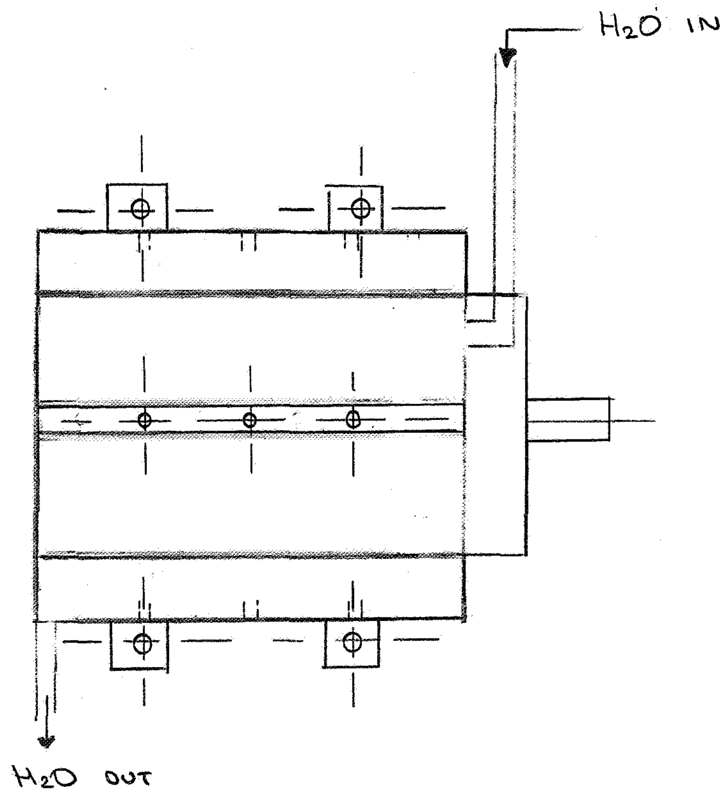


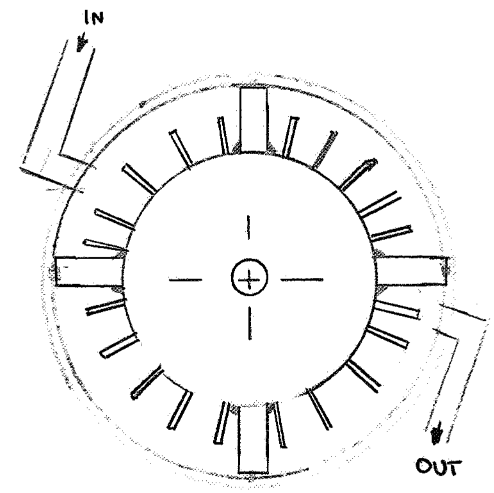
* 240VOLT, 40KW ALKALENE H_2-O_2 FUEL CELL BATTERY

DRAWING #20	MOTOR CASING (PROPOSED)	SCALE
	DRAWN BY: A.A DATE: 2/25	N/A

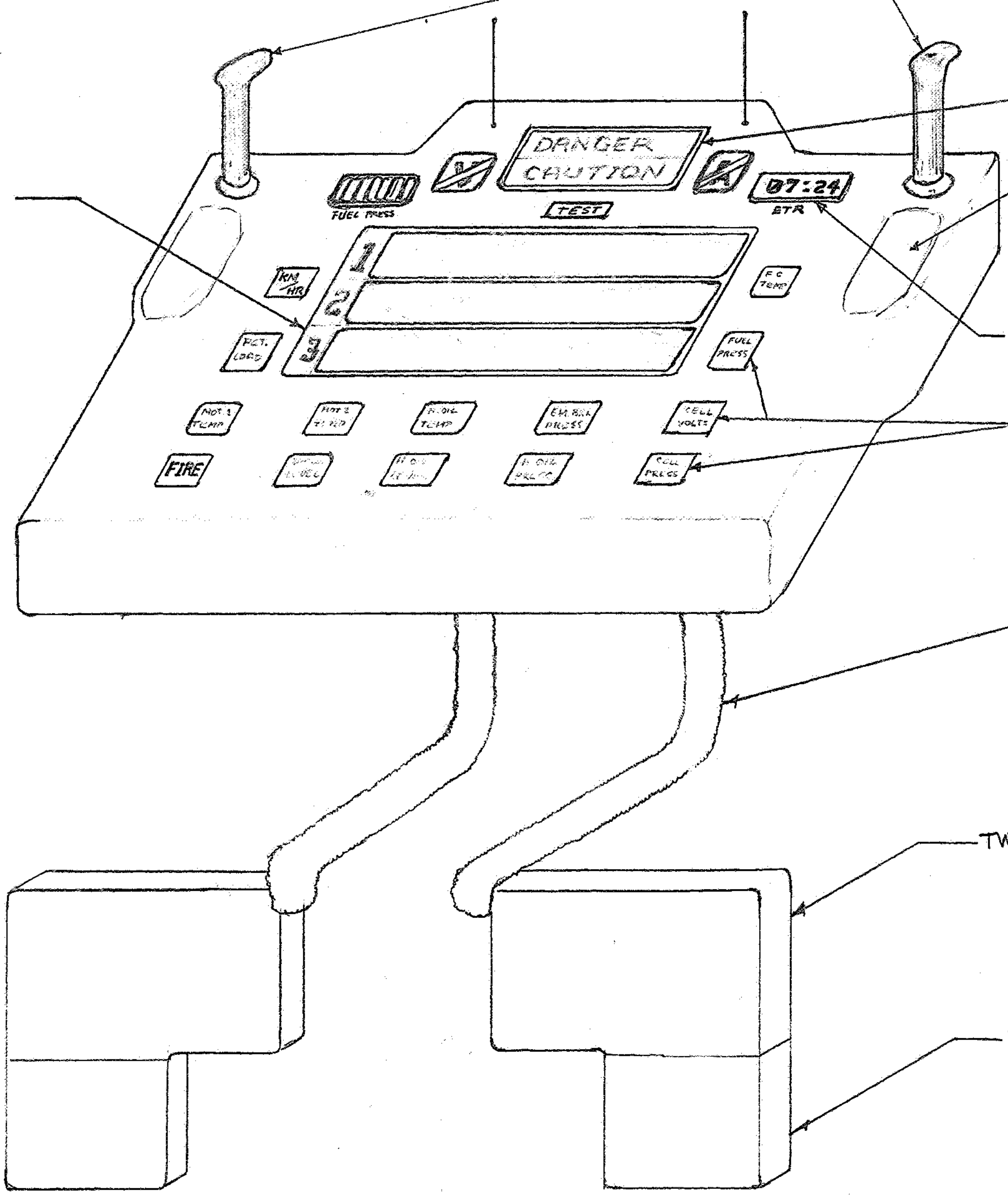


DRAWING #21	MOTOR INSIDE CASING SCEMATIC	SCALE N/A
	DRAWN BY: A. A DATE: 2/24	



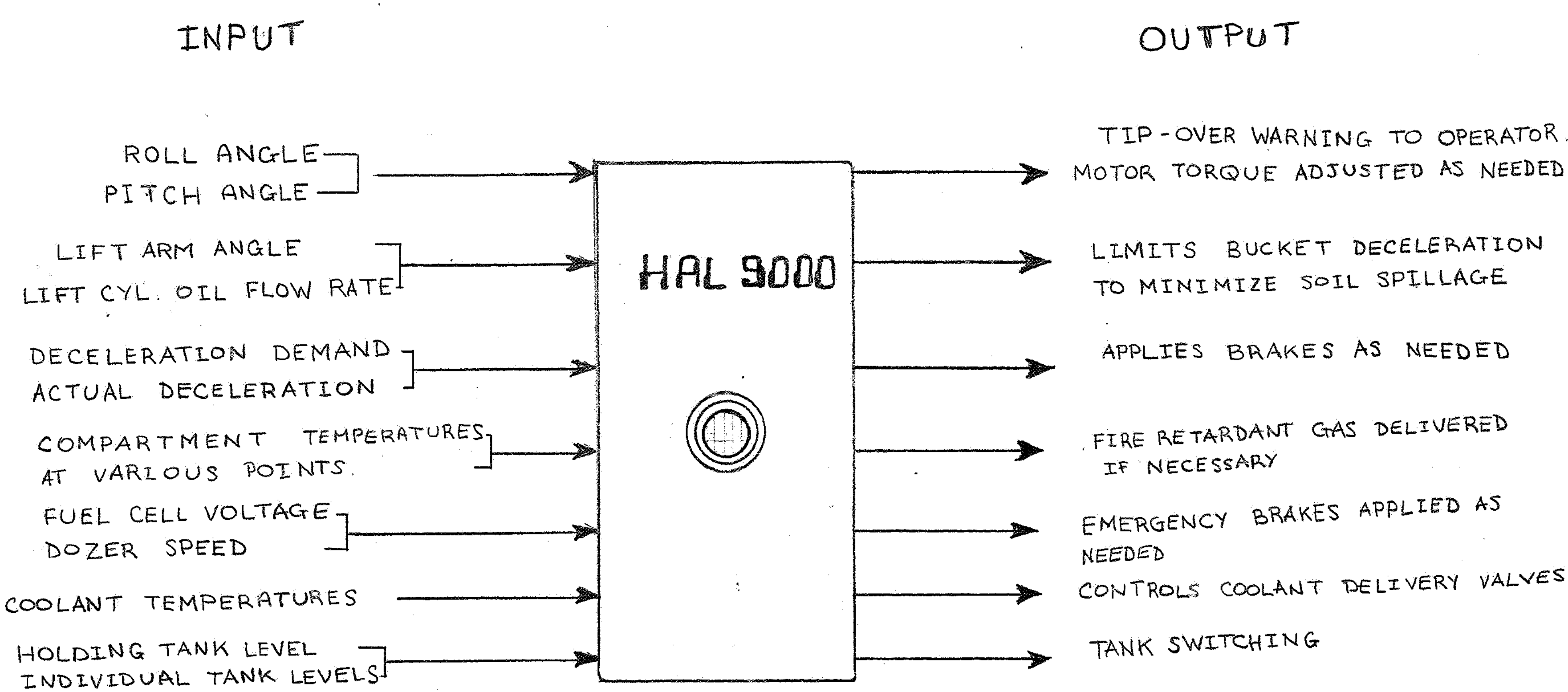


DRAWING #22	PROPOSED REMOTE CONTROL PANEL	SCALE
	DRAWN BY: C.M.T. DATE:	1:4

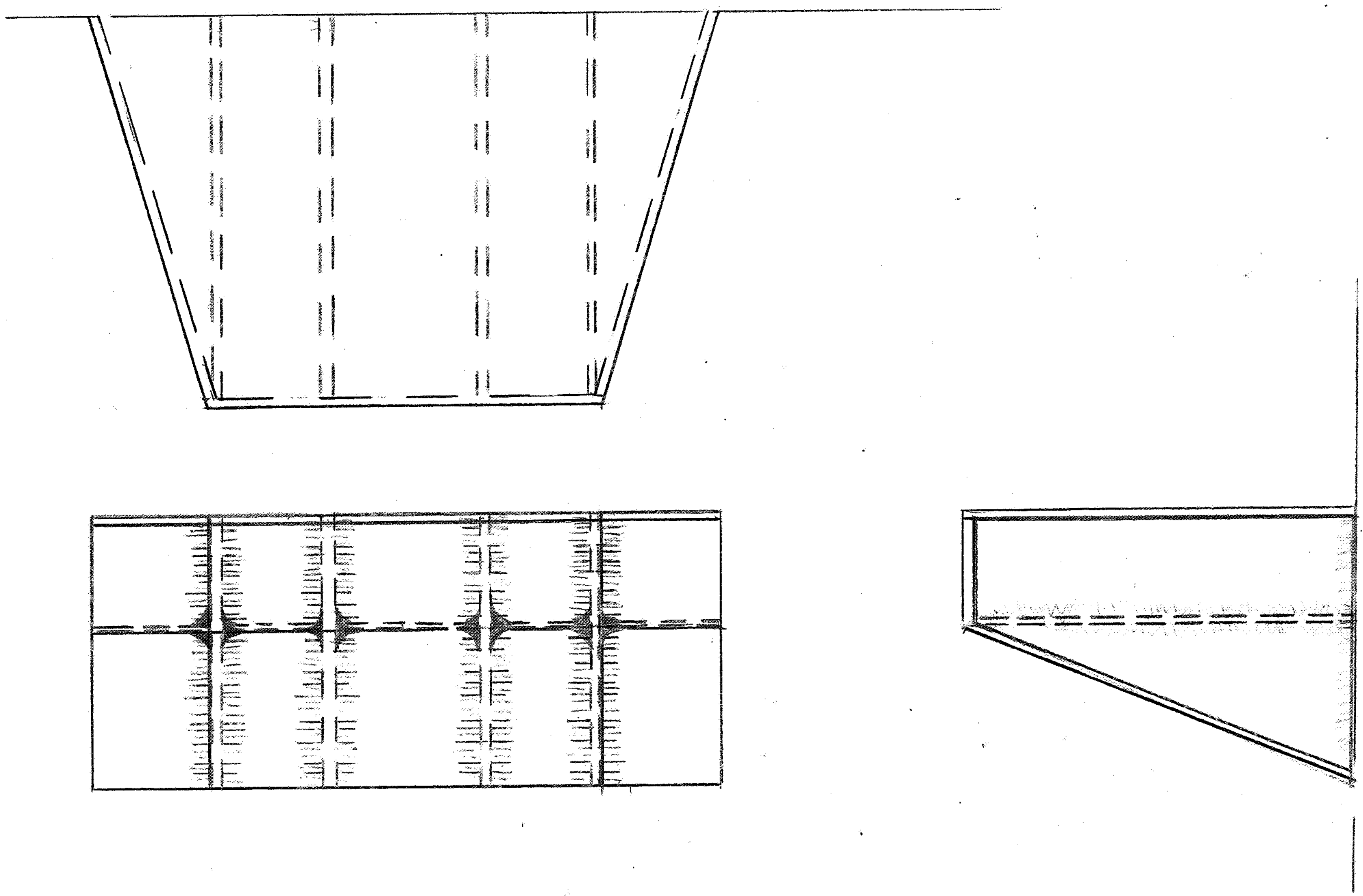


CASE MATERIAL: FIBERGLASS REINFORCED EPOXY LAMINATE.

NASA LUNAR DOZER DESIGN ENGINEERS	COMPUTER - AIDED CONTROL	NO.	DATE	SCALE	INIT.
		23	3/10/85	N/A	C.M.T.



NASA LUNAR DOZER DESIGN ENGINEERS	LIFT ARM MOUNTS	NO.	DATE	SCALE	INIT.
		24	10 MAR 85	4" = 1'-0"	N.M



ME 4182 WEEKLY PROGRESS REPORT

PERIOD: FEB. 20 THRU: FEB 26 TEAM NO.: TU4:30

TITLE: DESIGN OF A BULLDOZER APPROPRIATE FOR A LUNAR ENVIRONMENT.

COMMENTS:

THIS WEEK, OUR EFFORTS WERE MAINLY CONCENTRATED ON COMPLETING SECTION DRAWINGS AND SCHEMATICS. PATRICK THOLE AND JOHNÉ PARKER WROTE SEVERAL NEW STRESS ANALYSIS PROGRAMS FOR BULLDOZING APPLICATIONS, INCLUDING AN HP41CV PROGRAM WHICH DETERMINES THE MAXIMUM SAFE WEIGHT AND DRAWBAR PULL PER TIRE, GIVEN ASSUMED TIRE DIAMETER, WIDTH AND SINKAGE VALUES. SEVERAL EXISTING PROGRAMS ON BALL BEARINGS, TAPERED ROLLER (THRUST) BEARINGS, AND SILENT CHAINS WERE ALSO USED TO DETERMINE SHAFT AND GEAR DIAMETERS, BEARING LOAD RATINGS, AND CHAIN DIMENSIONS.

NAME, INITIALS	HOURS ENGINEERING	TECHNICIAN	CLERICAL	TOTAL
1) A. APOSTOLIDES	20			20
2) M. MANCUSO	20			20
3) N. McRAY	22			22
4) J. PARKER	18		3	21
5) P. THOLE	20			20
6) C. TOMLINSON	22			22
TOTALS =	122	—	3	125

ME 4182 WEEKLY PROGRESS REPORT

PERIOD: FEB. 13, 1985 THRU: FEB. 19, 1985 TEAM NO.: TU-4:30

TITLE: DESIGN OF A BULLDOZER APPROPRIATE FOR A LUNAR ENVIRONMENT

COMMENTS:

GROUP MEMBERS HAVE CONTINUED TO DO RESEARCH IN THEIR SPECIFIC AREAS; WE ARE NOW IN THE PROCESS OF WRITING UP OUR FINDINGS AND RECOMMENDATIONS IN EACH AREA. OUR RESEARCH INCLUDES A TOUR OF THE **CASE** FACILITIES IN ATLANTA; THIS WEEK, MARTIN MANCUSO, NELSON MCRAY AND JOANE PARKER EXAMINED THE CASE 1845B "UNI-LOADER" TO DETERMINE WHAT MODIFICATIONS NEED TO BE MADE.

NAME, INITIALS	HOURS ENGINEERING	TECHNICIAN	CLERICAL	TOTAL
1) A. APOSTOLIDES	2			
2) M. MANCUSO	5		1/2	
3) N. MCRAY	5			
4) J. PARKER	5			
5) P. THOLE	4			
6) C. TOMLINSON	4			
TOTALS =	25		1/2	25 1/2

WEEK 4- JANUARY 29, 1984

STATUS REPORT

We have researched a front-end attachment similar to existing designs. Several options were discussed as feasible solutions during our last meeting:

Hydraulic pistions
Gear Rack
Power Screws

Our group has divided into the following subsections:

Angelos Apostolides................Power source/Engine
Martin Mancuso......................Traction/Mobility
Nelson McRay........................Fuel/Energy Source
Johne' Parker.......................Cooling System
Patrick Thole.......................Power Train
Charles Thomlinson..................Materials/Structural Components

We have already begun research on these topics; at our next meeting we plan to discuss feasible alternatives.

M.E. 4182 MECHANICAL DESIGN ENGINEERING

PROJECT TITLE: DESIGN OF A BULLDOZER APPROPRIATE FOR A LUNAR ENVIRONMENT.

PROGRESS REPORT FOR THE PERIOD JAN. 23–29 TEAM: Tu-4:30

COMMENTS: PLEASE SEE ATTACHED SHEET.

GROUP MEMBERS:	HRS. WORK. COMPLETED THIS PERIOD:
1) ANGELOS APOSTOLIDES	5 hrs. (technical)
2) MARTIN MANCUSO	4 hrs. (technical)
3) NELSON MCRAY	5 hrs, technical; 1/2 hr, clerical
4) JOHNE' M. PARKER	5 hrs. (technical)
5) PATRICK THOLE	5 hrs. (technical)
6) CHARLES TOMLINSON	5 hrs. (technical)

* FOR A TOTAL OF 29 1/2 MAN-HRS. FOR THE PERIOD JAN. 23–29

In the process of researching the establishment of a permanent base on the lunar surface, NASA has recognized the need for soil-moving equipment. This equipment is vital to site preparation. Design requires that a working knowledge of the lunar environment and its effects on operation and product life of this equipment are attained.

ME 4182 WEEKLY PROGRESS REPORT

PERIOD: 3 - JAN. 16 THRU: JAN. 22 TEAM NO.: TU-4:30

TITLE: DESIGN OF A BULLDOZER APPROPRIATE FOR A LUNAR ENVIRONMENT.

COMMENTS:

OUR GROUP HAS MET SEVERAL TIMES TO DISCUSS PERTINENT DESIGN PARAMETERS; EA. GROUP MEMBER HAS ALSO DONE INDEPENDENT RESEARCH ON SUCH PARAMETERS AS:

A. SUITABLE MAT'LS: for protection against radiation, insulation, high strength/weight ratio.

B. SIZE & WEIGHT: approp. dimensions for space transport and use by astronaut.

C. TYPE OF POWER: I.C. Engine -vs- Batteries ⇒ Adv. and Disadvant of ea. process

D. TRACTION: Tracks -vs- Tires ⇒ Advantages & Disadvantages

E. ATTACHMENTS

F. MAINTENANCE & REPAIR

WE PLAN TO USE OUR KNOWLEDGE OF THE LUNAR ENVIRONMENT TO MODIFY EXISTING EARTH-MOVING EQUIPMENT SO THAT IT WILL FUNCTION IN A LUNAR ENVIRONMENT.

NAME, INITIALS	HOURS ENGINEERING	TECHNICIAN	CLERICAL	TOTAL
1) A. APOSTOLIDES	7	—	—	7
2) M. MANCUSO	7	—	—	7
3) N. McRAY	7	—	—	7
4) J. PARKER	7	—	—	7
5) P. THOLE	6	—	1	7
6) C. TOMLINSON	7	—	—	7
TOTALS =	41	-0-	1	42

** PBLM. STATEMENT ATTACHED

ME 4182 WEEKLY PROGRESS REPORT

PERIOD: FEB 27 THRU: MAR 5 TEAM NO.: TU 4:30

TITLE: Design of a bulldozer suitable for Lunar Applications

COMMENTS:

This week, the task for the final design was divided among the members. Nelson was responsible for the abstract, the introduction, the Power/Fuel consumption and the structural support. Martin was responsible for the final write up of the problem statement and for the wheels part of the drive train. Pat worked on performance objectives, the operating constraints, the transport specifications and the drive system. Charles did the brake system, the ergonomics/controls which include the computer interfaces and the control panel layout and also did the operating instructions. John undertook the task for selection of bearings, materials and coatings, and also did the hazard analysis, alternate designs and decision matrix. Finally, I was responsible for the motors and the cooling system.

NAME, INITIALS	HOURS ENGINEERING	TECHNICIAN	CLERICAL	TOTAL
1) A. APOSTOLIDES	1	3	11	15
2) M. MANCUSO	1	3	10	14
3) N. McRAY		2	12	14
4) J. PARKER		4	11	15
5) P. THOLE	1	6	7	14
6) C. TOMLINSON	2	4	8	14
TOTALS =	5	22	59	86

DESIGN OF A BULLDOZER APPROPRIATE FOR A LUNAR ENVIRONMENT

OUTLINE

I. ABSTRACT
II. PROBLEM STATEMENT
  A. BACKGROUND
    1. LUNAR ENVIRONMENT
      a. ATMOSPHERE
      b. TEMPERATURE
      c. GRAVITY
      d. SOIL CONDITIONS
      e. HEAT TRANSFER CHARACTERISTICS
    2. EXISTING EARTH - MOVING EQUIPMENT
  B. PERFORMANCE OBJECTIVES
    1. PAYLOAD
    2. DUTY CYCLE
    3. ATTACHMENTS
    4. MAINTENANCE
  C. CONSTRAINTS
    1. SPEED
    2. TRANSPORT SPECIFICATIONS
      a. DIMENSIONS
      b. WEIGHT
    3. POWER/FUEL CONSUMPTION
III. DETAILED DISCRIPTION
  A. POWER TRAIN
    1. MOTOR
    2. TRACTION DRIVE
  B. AUXILIARY ATTACHMENTS
    1. BLADE
    2. FRONT - END LOADER
  C. THERMAL SYSTEMS
  D. CHASSIS
  E. ERGONOMICS/CONTROLS
  F. MISCELLANEOUS
IV. HAZARD ANALYSIS
  A. FAILURE MODES AND CONSEQUENCES
    1. VEHICLE TIP OVER
    2. ABRUPT STOPS-SUIT PUNCTURE
    3. ATTACHMENT FAILURE
    4. FIRE/EXPLOSIONS
  B. FORESEEABLE USE AND MISUSE: EXCEEDING DEFINED DUTY CYLCE
V. OPERATING INSTRUCTIONS
VI. CONCLUSIONS AND RECOMMENDATIONS
VII. APPENDIX
    A. CALCULATIONS
    B. DRAWINGS
    C. ALTERNATE DESIGNS
    D. DECISION MATRIX
    E. WEEKLY PROGRESS REPORTS
    F. BIBLIOGRAPHY
    G. RECORD OF INVENTION FORMS
    H. COMPUTER USAGE

RECORD OF INVENTION - Part I

This is an important legal document. Read instructions carefully before filling in data.

PROJECT NO. ______

CONTRACT NO. ______

RECOMMENDED SECURITY CLASSIFICATION: secret

REC. OF INV. NO. ______

1. NAME OF INVENTOR: M. E. 4182 Tu4:30 Group *

POSITION: Sr. Mechanical Engr. Students

2. DEPARTMENT OR DIVISION: GEORGIA INSTITUTE OF TECHNOLOGY

3. DATES OF EMPLOYMENT: Wtr. Qtr. 1985 (January 4 - March 12)

4. PRESENT ADDRESS (No. Street, City, County, State): SSTC #1 Room 218, Atlanta, GEORGIA

TELEPHONE: (404) 894-3218

PERMANENT OR UNTIL

5. PERMANENT ADDRESS (No. Street, City, County, State)

TELEPHONE

6. NAMES (S) AND ADDRESS (ES) OF CO-INVENTORS (If any)

* Angelos Apostolides, Martin Mancuso, Nelson McRay, Johne' Parker Patrick Thole, and Charles Tomlinson.

7. DESCRIPTIVE TITLE OF INVENTION

The Lunar Dozer

8. LIST DRAWINGS, SKETCHES, PHOTOS, REPORTS, DESCRIPTIONS, NOTEBOOK ENTRIES, ETC. WHICH SHOW OR DESCRIBE INVENTION

See the DRAWINGS section of the Appendix in the preceding report.

9. EARLIEST DATA AND PLACE INVENTION WAS CONCEIVED (Brief outline of circumstances)

January 8, 1985

10. DATE AND PLACE OF FIRST SKETCH, DRAWING OR PHOTO

February 12, 1985

11. DATE AND PLACE OF FIRST WRITTEN DESCRIPTION

February 26, 1985

12. DISCLOSURE OF INVENTION TO OTHERS

NAME, TITLE AND ADDRESS	FORM OF DISCLOSURE	DATE AND PLACE OF DISCLOSURE	WAS SIGNATURE OBTAINED (YES OR NO)
Mr. J. W. Brazell, M. E. Instructor	oral		
GEORGIA INST. OF TECHNOLOGY			

12.A IMPORTANT - HAVE ANY PUBLICATIONS OR REPORTS BEEN MADE ON THIS INVENTION?

yes; see preceding report.

13. DATE AND PLACE OF COMPLETION OF FIRST OPERATING MODEL OR FULL SIZE DEVICE

n/a

14. PRESENT LOCATION OF MODEL

15. DATE, PLACE, DESCRIPTION AND RESULTS OF FIRST TEST OR OPERATION

16. NAMES AND ADDRESSES OF WITNESSES OF FIRST TEST

17. DATE, PLACE, DESCRIPTION AND RESULTS OF LATER TESTS (name witnesses)

18. IDENTIFY RECORDS OF TESTS AND GIVE PRESENT LOCATION OF RECORDS

19. PRIOR REPORTS OR RECORDS OF INVENTION TO WHICH INVENTION IS RELATED

none

20. OTHER KNOWN CLOSELY RELATED PATENTS, PATENT APPLICATIONS AND PUBLICATIONS

PATENT OR APPLICATION NO.	DATE	TITLE OF INVENTION OR PUBLISHED ARTICLE	NAME OF PUBLICATION
none			

21. EXTENT OF USE: PAST, PRESENT AND CONTEMPLATED (Give dates, places and other pertinent details) The lunar dozer will be used to move lunar dust and rock away from the permanent space station that NASA wishes to build on the moon.

22. DETAILS OF INVENTION HAVE BEEN RELEASED TO THE FOLLOWING COMPANIES OR ACTIVITIES

NAME AND ADDRESS	INDIVIDUAL OR REPRESENTATIVE	CONTRACT NO.	DATE
NASA, Huntsville, AL			

SIGNATURE OF INVENTOR

DATE

RECORD OF INVENTION - Part II

(Attach to Record of Invention Part I)

REC. OF INV. NO. ______________

This Disclosure of Invention should be written up in the inventor's own words and generally should follow the outline given below. Sketches, prints, photos and other illustrations as well as reports of any nature in which the invention is referred to, if available, should form a part of this disclosure and reference can be made thereto in the description of construction and operation.

1. INVENTORS NAME (S) M. E. 4182 Design Group Tu4:30

2. TITLE OF INVENTION The Lunar Dozer

For answers to following questions use remainder of sheet and attach extra sheets if necessary.

3. GENERAL PURPOSE OF INVENTION. STATE IN GENERAL TERMS THE OBJECTS OF THE INVENTION.
4. DESCRIBE OLD METHOD(S) IF ANY, OF PERFORMING THE FUNCTION OF THE INVENTION.
5. INDICATE THE DISADVANTAGES OF THE OLD MEANS OR DEVICE(S).
6. DESCRIBE THE CONSTRUCTION OF YOUR INVENTION, SHOWING THE CHANGES, ADDITIONS AND IMPROVEMENTS OVER THE OLD MEANS OR DEVICES
7. GIVE DETAILS OF THE OPERATION IF NOT ALREADY DESCRIBED UNDER 6.
8. STATE THE ADVANTAGES OF YOUR INVENTION OVER WHAT HAS BEEN DONE BEFORE.
9. INDICATE ANY ALTERNATE METHODS OF CONSTRUCTION.
10. IF A JOINT INVENTION, INDICATE WHAT CONTRIBUTION WAS MADE BY EACH INVENTOR.
11. FEATURES WHICH ARE BELIEVED TO BE NEW.
12. AFTER THE DISCLOSURE IS PREPARED, IT SHOULD BE SIGNED BY THE INVENTOR(S), AND THEN READ AND SIGNED AT THE BOTTOM OF EACH PAGE BY TWO WITNESSES USING THE FOLLOWING STATEMENT:

"DISCLOSED TO AND UNDERSTOOD BY ME THIS______DAY OF_________19___
SIGNATURE______________________________."

3. Many tools and vehicles have been used on the moon; however, none of these was designed for long-term use or soil-moving capabilities. NASA wishes to build a permanent space station on the moon; our bulldozer will be used to move lunar dust and rocks away from this site. Since there are no exising space vehicles upon which to base a bulldozing design, we have modified existing earth-moving equipment so that it will operate in a lunar environment.

Questions 4 - 6 do not apply to this design.

7. Please see the section on OPERATING INSTRUCTIONS in the preceding report.

REC. OF
INV. NO. ____________

8. This question does not apply.

9. Please see Appendix: ALTERNATE DESIGNS in the preceding report.

10. Please see PROGRESS REPORTS in the Appendix.

11. The design basically modifies an existing uni-loader for lunar applications; therefore, no features of this machine are totally new (although the synthesis of our ideas led to the design of a unique machine).

INVENTOR

March 12 1985

DISCLOSED TO AND UNDERSTOOD BY ME ON THIS 12 DAY OF March 1985

John M. Parker

WITNESS

DISCLOSED TO AND UNDERSTOOD BY ME ON THIS 12 DAY OF March 1985

Patrick D. Thole

WITNESS

10.7. Bibliography:

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